OFFICIAL
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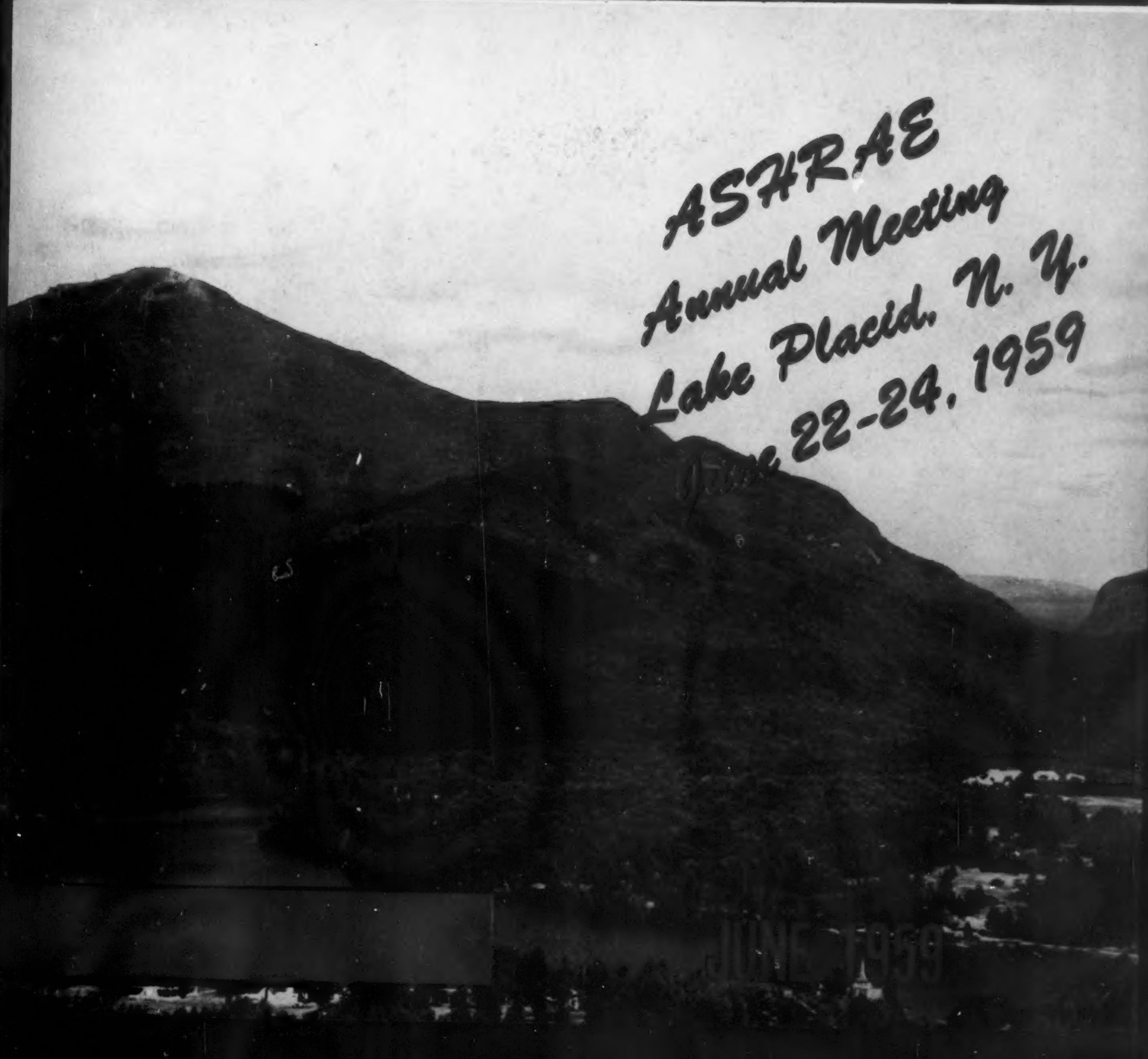
ASHRAE JOURNAL

Heating • Refrigerating • Air Conditioning • Ventilating

AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS



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Comment

THERE MUST BE A REASON

Why people, individually or in the mass, do certain things at the times when they do them must ever remain one of the more inscrutable mysteries; despite the efforts of politicians, psychologists, psychoanalysts, educators, pitchmen and the hidden persuaders to explore the matter. Why did Simple Simon do his fishing in a tub of water? Why did Sir Oliver Lodge become beguiled by the mysteries of ectoplasm? Why did millions give their lives and blood in contention over the right of a state to withdraw from the Union? Why should ideologies matter to anyone?

There is a matter of societies. Why do people join them? As a professional engineering society, the American Society of Heating, Refrigerating and Air-Conditioning Engineers exists ". . . to advance the arts and sciences . . . for the benefit of the general public . . . Matters pertaining to politics, religion or solely to trade shall not be discussed at any meeting of the Society, nor be included in any of its publications". We quote the By-laws.

Yet, beyond all question, there are many members of ASHRAE who do belong for non-engineering, non-scientific reasons and who expect to gain from their membership certain values unallied with the realms of a professional society; just as there are those whose perspective is non-commercial and just as the interests of both groups may be partially social. The only hazard here entailed, but it is a real one, is that the interests and anticipations, for things present or things to come, of the non-engineering group may conflict with what must be the basic program.

Let us not be misunderstood. Ivory tower engineering alone gets no one anywhere. Equipment has to be promoted, marketed and sold. As long as it is accepted that engineering is the indispensable base, there is no problem; but should pressures for non-engineering services and meetings mount, watch out.

We think that societies, like other things, are vulnerable to change, that only continual upgrading will maintain high standards, that strength is the outcome of largely undiluted concepts and that each and every member should be under no illusions as to what it is all about.

Edward R. Searles
Editor

Late news highlights

Separated In order to permit more effective dissemination of its findings to science and industry and to meet the specialized needs of today's scientists, engineers and mathematicians, the National Bureau of Standards will publish its Journal of Research in four separate sections, beginning July 1, corresponding to subject matter fields. The four sections (available in individual subscriptions) are: Section A – Physics and Chemistry; Section B – Mathematics and Mathematical Physics; Section C – Engineering and Instrumentation; and Section D – Radio Propagation.

Comprehensive Directory Basic information regarding international, national, regional, state and local engineering groups is contained in the 1959 Engineering Societies Directory published by Engineers Joint Council. Also included are address, membership, secretary and organizational publications. Directory is available from EJC, 29 West 39th Street, New York 18, N. Y., at $3.50.

Thermodynamics Both physical and mathematical aspects of thermodynamics are covered in the text, **Thermodynamics**, by Gordon J. Van Wylen. Written from an engineering perspective, the volume defines basic laws and terms. First law of thermodynamics, including flow into an open system and conservation of mass, and second law are developed. Also covered are processes in reciprocating machinery and cycles; power and refrigeration cycles; one-dimensional flow through nozzles and orifices; and axial flow through blade passages. (John Wiley & Sons, New York. 567 pages, $7.95.)

Cryogenic distillation Separation of hydrogen isotopes has been accomplished at the National Bureau of Standards' Cryogenic Engineering Laboratory (Boulder, Colo.) in a pilot plant capable of distilling liquid hydrogen by fractionation, yielding specific amounts of pure liquid deuterium and deuterium-free hydrogen. This method may provide a means for industry to produce large quantities of deuterium.

How to do it "Warm Air Perimeter Heating as Applied to Structures with or without Basements" contains revised standards to clarify ways to install a perimeter warm air system in a concrete slab. Five type classifications of ducting material to be used in concrete slab construction are included together with accepted design methods and proper installation requirements. Copies are available from the National Warm Air Heating and Air Conditioning Association, 640 Engineers Building, Cleveland 14, Ohio.

George A. Horne dies Just as we go to press we learn of the death of George A. Horne, Past President and Honorary Member of ASRE. Mr. Horne joined the Society in 1911, becoming President in 1924. In 1944 he was made an Honorary Member, the fifth. Further details of Mr. Horne's career and contributions will appear in our July issue.

Good influence "Work output of employees in an air conditioned space is greater, on an average, by more than nine percent than that of employees in similar space which is not air conditioned" is the summary of a five-month study by the General Services Administration of the Public Building Service, Washington 25, D. C. Other advantages of air conditioning cited in the report are: decrease in absenteeism and contribution to employee comfort and morale.

Treatise "An Introduction to Physical Microclimatology" by F. A. Brooks discusses heat vapor exchange close to ground and provides working knowledge of the thermal relationships between the solar and nocturnal radiation and resulting equilibrium temperatures in our climatic environment. A limited edition of this treatise has been prepared by the Department of Agricultural Engineering, College of Agriculture, Agricultural Experiment Station, University of California, Davis, Calif.

Revised edition Completely brought up to date, Modern Air Conditioning, Heating and Ventilating, Third Edition, by Willis H. Carrier, Realto E. Cherne, Walter A. Grant and William H. Roberts incorporates new developments and latest information and data. It includes recent material on comfort and heat transfer; heat pumps; air condensers; water treatment; high-velocity air duct design; psychrometric processes; cooling coil selection and rating; and control devices. (Pitman Publishing Corp., New York. 592 pages, $12.)

Code is law Detroit, Mich., has adopted as law the American Standard Safety Code for Mechanical Refrigeration, ASA B9.1–1958 and it is included in the Detroit Refrigeration Code Book. Other neighboring communities have also adopted the B9 Code as law.

Reasons for decline Freshman engineering enrollment has declined markedly for the first time in eight years, a survey by the Engineering Manpower Commission in cooperation with the American Society for Engineering Education reveals. Heads of engineering schools cited these reasons for the declining enrollment: false appraisals of the long range engineering career opportunities by counsellors, students and parents; increased concern about rigors of engineering curriculum; and increased interest by potential engineering students in other scientific fields.

Annual report "Research on the Refrigeration of Perishable Commodities," annual report of the Director of the Refrigeration Research Foundation, 12 North Meade Avenue, Colorado Springs, Colo., describes the 21 studies the Foundation is sponsoring in its current research program and lists its publications for 1958.

Criteria for ducts A special Advisory Committee of the Building Research Advisory Board, headed by Paul R. Achenbach, Chief of the Air Conditioning, Heating and Refrigeration Section, National Bureau of Standards, has prepared a report on "Criteria for Ducts to be Used in Residential Warm Air Heating and Air Conditioning Systems." Objective of the study was to define properties or characteristics a duct should possess to perform satisfactorily and to describe tests needed to demonstrate these qualities. As a result of the study recommendations were made to serve as a basis for making acceptability determinations based on performance for ducts fabricated from any material. Copies (NAS-NRC Publication 651) are available from the Printing and Publication Office, National Academy of Sciences–National Research Council, 2101 Constitution Avenue, Washington 25, D. C., for $1.25.

Higher incomes for teachers According to a survey published for Engineers Joint Council, 29 West 39th Street, New York 18, N. Y., the average professional income of engineering teachers in the United States has risen 8.3% since 1956 and basic teaching salaries have increased 13.5% over the two-year period. Survey covered more than 5,000 engineering teachers.

NOTICE TO MEMBERS OF 1959 ANNUAL MEETING

The Annual Meeting of the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. will convene at Lake Placid, N. Y. at 9:00 a.m., Monday, June 22, 1959.

Continuing through Wednesday, June 24, the meeting will include eight Technical Sessions covering Compressors, Ventilation, Noise, Heat Transfer and General Refrigeration and Air Conditioning. There will be a Domestic Refrigerator Engineering Conference, Industrial Ventilation Conference and a Conference on Cryogenics.

A brief business session during which Officers and committees will present reports is scheduled for Monday, June 22.

Every member should plan to attend the Lake Placid Meeting.

CECIL BOLING
President

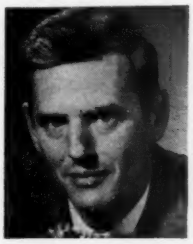
A. V. HUTCHINSON
Executive Secretary

ASHRAE JOURNAL

JUNE 1959

Convective films

evaluated for wire and tube heat exchangers



O. W. WITZELL
Member ASHRAE

W. E. FONTAINE
Member ASHRAE

W. J. PAPANEK

The initial phase of this study was developed to determine convective film coefficients in natural convection for horizontal exchangers.[1] The attack depended on the measurement of the total heat flow from the exchanger by determination of flow rate and temperature drop of water passing through the exchanger in an environment of known magnitude.

The data obtained by the equipment designed to measure some of the variables were questionable. It became apparent that the contribution to total heat transfer by radiation was significant. Measurement techniques devised to obtain the radiative contribution were of such character that it was impossible to obtain the degree of accuracy desired without some modification. In addition, the convective heat transfer was affected by fluctuating environmental conditions. Improved techniques for the measurement of temperatures and mass flow rates were devised with the use of instrumentation that provided a greater degree of accuracy.

In order to eliminate the variable conditions in the environment from the viewpoint of both convection and radiation, the exchanger was installed in a cell that was designed to provide the necessary control of the environment. The cell was of such nature that convection air currents were stable and the wall temperatures could be held constant. In addition, the cell design permitted measurements to be made when the heat exchanger was in various positions other than horizontal.

DISCUSSION OF APPARATUS

The apparatus is similar to that described in a previous paper[1] except for some important modifications to allow more precise determination of the variables affecting the heat flow. The design is such as to allow the determination of the convective film coefficient on the outside of the exchanger by measuring the total heat flow and subtracting the contribution due to radiation. The exchanger was enclosed in a test cell which had been covered with a dull black paint to allow precise determination of the surface emissivities. The wall temperature of the enclosure was determined from thermocouples placed at various locations on the inside wall of the enclosure.

The connections to the exchanger and the thermocouple wire harness were so arranged that the exchanger could be adjusted at various angles in the enclosure.

In order to determine the ambient air temperature in the immediate vicinity of the exchanger, thermocouple readings were taken around the exchanger to determine the extent of the air disturbance due to convective currents. These readings indicated the minimum distance between the ambient thermocouples and the exchanger. The ambient temperature of the air was obtained from thermocouples mounted on a tree in the test cell. Thus, a continuous indication of the ambient air temperature from top to bottom of the cell as well as the wall temperature distribution was available to determine the existence of a steady state condition before heat transfer data were obtained.

O. W. Witzell and Professor W. E. Fontaine are with the School of Mechanical Engineering, Purdue University. W. J. Papanek is with Lockheed Missiles System. This paper was presented as "An Evaluation of the Convective Film Coefficients for Wire and Tube Heat Exchangers" at the 45th Semiannual Meeting of ASRE in New Orleans, La., December 1-3, 1958.

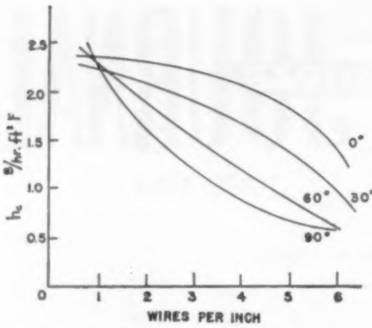


Fig. 1 Variation in the convective film coefficient with number of wires and the angle of inclination with the horizontal

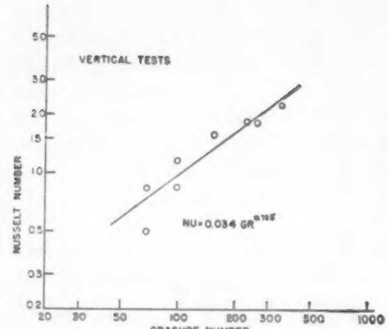
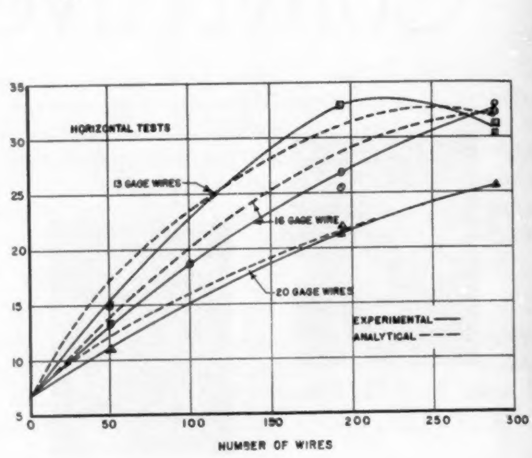


Fig. 2 Correlations among the Nusselt number and Grashof number for heat exchangers in the vertical position

Fig. 3 Heat transfer as a function of the number of wires and wire gages



Measurements of heat transfer from some exchangers were made with the tubes at various angles but with horizontal wires. The convective coefficients for these conditions were computed and the results are shown in Fig. 1. The effect of interference between the wires is clearly evident. Interference becomes more important as the number of wires are increased and as the angle from the horizontal is increased.

The data and calculated results obtained for the vertical tests are shown in Fig. 2. The chart shows a typical Nusselt-Grashof correlation in which the heat exchanger was considered to be acting as a flat plate.

The data for the horizontal condition represent a careful survey of the operating characteristics of the heat exchanger. Fig. 3 shows the effect of number of wires and wire size on the heat transfer from the exchanger. The experimental information indicates the apparent drop-off of heat transfer as the ratio of wire spacing to wire size becomes small. This effect is shown markedly in the case of the 13 gage wires where a point is reached at which the addition of extra wires actually produces a decrease in the heat transfer.

The decreasing heat flow obtained by the addition of extra wires in the case of 16 and 20 gage, tends to indicate a maximum heat flow might exist with close wire spacing for these wire arrangements.

In order to generalize the information obtained for the horizontal tests, it becomes necessary to describe the geometry of the heat exchanger. In general, correlations of the heat transfer coefficient of cylinders in natural convection in the same medium and at about the same temperature are given by the equation,

$$h = KD^{-1/4} \qquad (1)$$

where K is a constant.

The total heat transfer from wires and tubes is

$$Q_{total} = Q_w + Q_t = h_{tot}A_{tot}\Theta_t \qquad (2)$$

where

$$Q_w = h_wA_w\Theta_w$$
$$Q_t = h_tA_t\Theta_t$$

so that

$$h_{tot}A_{tot}\Theta_t = h_tA_t\Theta_t + h_wA_w\Theta_t\eta \qquad (3)$$

and

$$h_{tot} = \frac{A_th_t + \eta A_wh_w}{A_t + A_w} \qquad (4)$$

Substitution from equation (1) gives

$$D_{char}^{-1/4} = \frac{A_tD_t^{-1/4} + \eta A_wD_w^{-1/4}}{A_t + A_w} \qquad (5)$$

and

$$D_c = \left[\frac{A_t + A_w}{\dfrac{A_t}{D_t^{1/4}} + \dfrac{\eta A_w}{D_w^{1/4}}} \right]^4 \qquad (6)$$

This characteristic diameter is used to compute the Nusselt and Grashof numbers for the complete exchanger.

In view of the overlapping boundaries indicated by Fig. 4, it is necessary to modify the general correlation for heat flow from cylinders by a geometry factor. Since the tubes play only a small part in the total heat transfer from the exchanger, the geometry factor should be primarily dependent on the wire configuration alone. One such correlation is

$$N_{Nu} = [N_{Gr}]^a \left[\frac{S_w - D_w}{S_w} \right]^b \qquad (7)$$

The constant b was so chosen as to best represent the data shown graphically in Fig. 3. The numerical value of the constants c and a were obtained from the graph of the data as shown in Fig. 5 so that the final form of the equation becomes

$$N_{Nu} = .905\ [N_{Gr}]^{.376} \left[\frac{S - D}{S} \right]^{1.2} \qquad (8)$$

Equation (8) represents the generalized correlation for the horizontal condition for wire and tube heat exchangers in natural convection. In order to test its ability to predict operating conditions, values of heat flow were computed for some representative conditions obtained in the experiments. Fig. 3

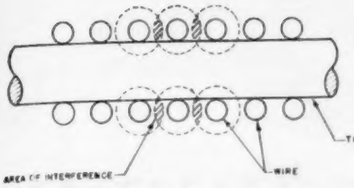


Fig. 4 Boundary layer interference

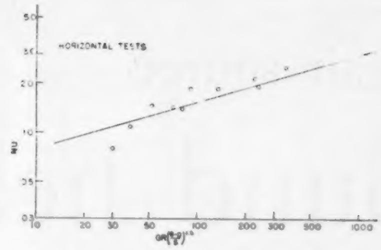


Fig. 5 Correlation of the Nusselt number and Grashof number for heat exchangers in horizontal position

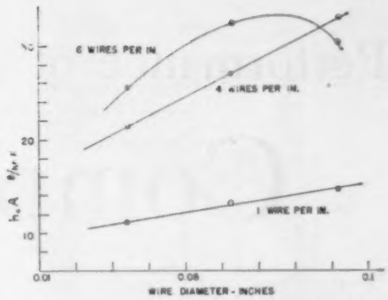


Fig. 6 Heat transfer characteristics as a function of the wire diameter and the number of wires

shows the result of such calculations and compares the experimental determinations of hA as contrasted to the values computed from the equation. The agreement is satisfactory since the calculated information shows the tendency toward maximum heat transfer indicated by the experimental data.

DISCUSSION OF RESULTS

It has been stated in a previous paper that for the conditions in the experiments as described above the Prandtl number is essentially constant and the relationship

$$N_{Nu} = f(N_{Gr}) \qquad (9)$$

is valid.

The total heat transferred from the heat exchanger was calculated by the relationship

$$q = Mc_p(t_{in} - t_{out}) \qquad (10)$$

where M and Δt are measurable quantities. When the heat transferred by radiation is subtracted from total q, then the remainder is the heat that is transferred by convection, thus

$$q_c = q_t - q_r \qquad (11)$$

When q_c is known then the convective film coefficient h can be computed by the equation

$$q_c = Ah_c\theta \qquad (12)$$

where θ is the temperature difference between the bulk fluid flowing and the ambient air. After having computed h_c, the Nusselt number $\left(\frac{hD_c}{K}\right)$ can be determined since the characteristic dimension (D_c) and the thermal conductivity K are known. The Grashof number can be computed directly since all of the variables are known. From this information a plot, N_{Nu} vs N_{Gr} can be made. For information referring to horizontal tests, see Fig. 5. From this plot a generalized relationship can be made as indicated by equation (8). It should be emphasized that this equation is valid for any temperature difference and is independent of heat exchanger geometry; i.e., wire diameter, wire spacing, and overall dimensions.

Since $D_c = f(D_w, S_w \text{------})$
Where D_w = wire diameter
S_w = wire spacing

definite selection of these variables, when used with the plot in Fig. 5 will result in a specific value of h_c for this design criteria. This information can then be used in determining the plot hA vs the number of wires—see Fig. 3. In this plot the solid lines represent the experimental data and the dotted lines represent the curves as calculated in the manner described above.

This plot is particularly significant since it indicates that the generalized relationship as described by equation (8) is valid since it approximates the experimental curve within reasonable limits. Extended tests and greater amounts of data will probably lead to bringing these curves more closely together. At any rate, the data and experimental procedure verify the calculated plot within reasonable limits of accuracy.

Fig. 3 is a plot showing how the convective film coefficient h varies with the number of wires for definite values of the angle of inclination with the horizontal.

Fig. 6 is a plot showing the relationship between the product hA and the wire diameter with the number of wires as the parameter. In this plot hA was chosen since it is directly proportional to q_c.

These plots, then, Figs. 1, 3, 5, and 6 are useful in the design of a particular heat exchanger when the amount of heat to be transferred and the temperature difference are known. All other variables such as wire size, wire spacing, tube spacing, and tube diameter can be selected on the basis of economic and manufacturing criteria. The value of these plots is that in every case they indicate the change in h_c when the number of wires, temperature difference, and the wire diameter are the variables.

Fig. 2 is a plot that shows the relationship between the Nusselt number and Grashof number for a heat exchanger in the vertical position. In this case, experiments indicated that the wire on tube heat exchanger acted as a vertical flat plate. This plot can be used for design in the same manner as described above for heat exchangers in the horizontal position.

CONCLUSIONS

The present work has resulted in a correlation among N_{Nu} and N_{Gr} and a resultant generalized equation which is not restricted by heat exchanger geometry. Either the plot or the generalized equation can be used in the determination of the convective film coefficient h_c for any physical situation regardless of heat exchanger geometry.

Future work should be directed to determine more information on the heat transfer characteristics when the exchanger is in a position other than horizontal. More data are needed in order to establish a generalized equation of greater accuracy, and finally the work should be extended to include forced convection.

REFERENCES

1. *Heat Transfer Characteristics of Wire and Tube Condensers,* Witzell, O. W., and Fontaine, W. E., Refrigerating Engineering, Vol. 65, No. 3, March 1957, pp. 33-37
2. *Design of Wire and Tube Condensers,* Witzell, O. W., and Fontaine, W. E., Refrigerating Engineering, Vol. 65, No. 9, September 1957, pp. 41-44
3. Thesis, Master of Science in Mechanical Engineering, Purdue University, Lafayette, Indiana, January 1958, Papanek, W. J.

Performance of an air source

Compound heat pump

A. D. SPILLMAN

Air appears to be the most promising future source of heat for heat pump installations, limited only by the availability of dependable equipment to give efficient year-round heating and cooling comfort. Recently, compound or multi-stage compression machines have been adapted to operate at reduced compression ratios, with increased output, and higher performance efficiencies. With these, auxiliary resistance heaters are not necessary. These developments prompted an investigation of the heat pump as a suitable means of heating and cooling a specific Philadelphia office building.

Heating installations in the Philadelphia area are designed for an outdoor temperature of 0 F, and in order to extract heat from the outside air at that temperature, it is necessary that the refrigerant in the evaporator of the heat pump be at a temperature of about –30 F to accomplish a rapid heat transfer. On the condensing side of the heat pump the refrigerant must be at a temperature of about 120 F in order to provide 115 F water for circulation as the heating medium throughout a building. As the temperature of the refrigerant gas is a function of its pressure, it follows that to effect a temperature change of from –30 to +120 F the compressor would require a high compression ratio; where the compound compression machine offers advantages.

After several interviews with air-conditioning equipment and utility representatives, the new building owner requested that proposals be submitted on furnishing and installing a heat pump, and on the cost of electric power in connection with it. In addition, studies were made of the same heating and cooling requirements with fuel-fired boilers for heating, and a mechanical system of air conditioning for cooling.

The building is a two-story structure, of approximately 15,840 sq ft area. The sidewalls are common party walls, for the most part, with relatively small heat gain or heat loss from these areas. The front of the building is mostly glass, and in the rear the percentage of glass area is relatively high. Insulation on other exposed areas was carefully planned for economy.

Little information was available on the use of the multi-stage compression machine for year-round operation, but calculations indicated that two 60 hp machines would provide more than adequate capacity for heating; and further, that the installation of two units would provide reserve capacity, as the operation of one unit would supply the demands of building heat most of the time.

Heat pump equipment was installed in the partial basement, with an outdoor air cooler-con-

TABLE I

HEATING COSTS — ESTIMATED COMPARED WITH ACTUAL

Month	Estimated[a]	Actual[b]	Normal Degree-Days
January	$ 395.00	$ 416.00	933
February	352.00	324.00	837
March	284.00	211.00	667
April	158.00	147.00	369
May	40.00	27.00	93
June			
July			
August	COOLING ONLY		33
September			
October	94.00	54.00	219
November	221.00	212.00	516
December	361.00	343.00	856
	$1,905.00	$1,734.00	4,523

[a] Includes operation of all electric auxiliaries used for heating.
[b] Actual heat pump operating costs prorated to a normal degree-day basis of 4523 for Philadelphia. Includes operation of all electric auxiliaries used for heating.

A. D. Spillman is Assistant Manager, Sales Applications Dept., Philadelphia Electric Co. This paper was presented at the Symposium on Heat Pump Performance at the 65th Annual Meeting of ASHAE, January 26-29, 1959.

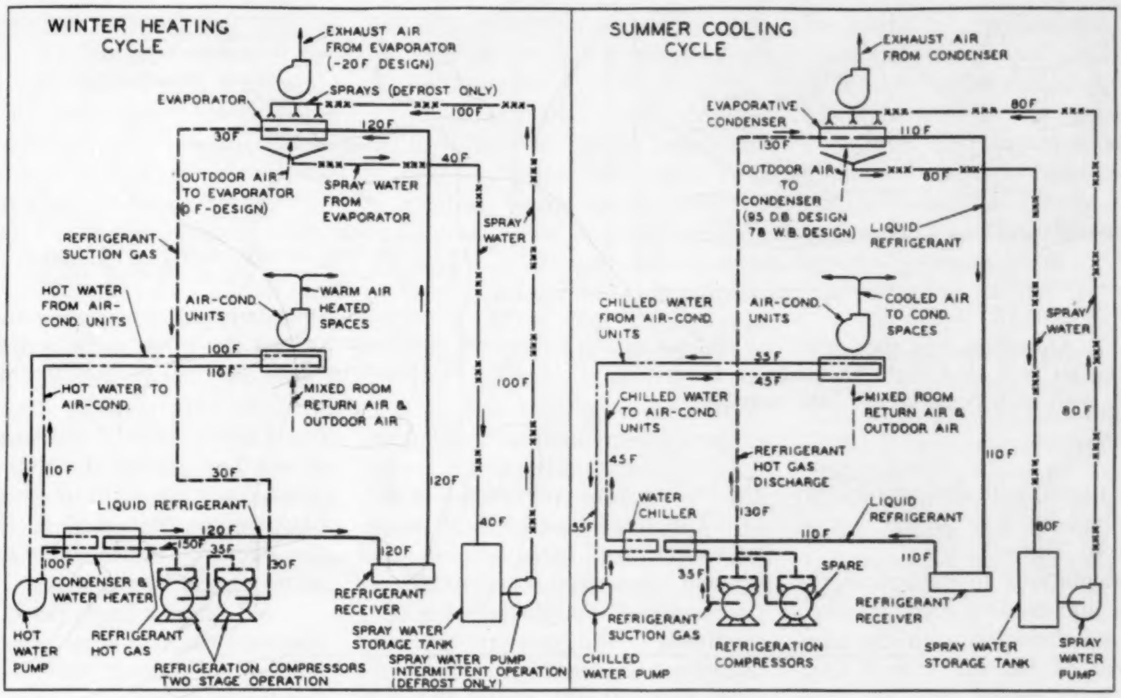


Fig. 1 Simplified piping diagrams of summer cooling and winter heating cycles

denser on the building roof. This installation utilizes air as the heat source, and water as the distributing medium for heating or cooling. The water circulates through a closed system of three indoor-air-handling units, totaling 20,800 cfm, each serving a separate zone. Concealed ducts and ceiling type diffusers supply the large drafting area with air of the desired temperature. The private offices are handled independently with floor mounted fan-coil room units, equipped with combination hot and chilled water coils.

The heat pump system is designed for automatic operation to provide 45 F water for summer cooling and 115 F water for winter heating. Change-over from cooling to heating is accomplished through a simple control switch.

Fig. 1 is a simplified diagram of the complete cycle of heating and cooling. It is basically an air-to-water-to-air system, which in the heating season simply takes heat from the air, transfers it from the refrigerant to water in a heat exchanger, which in turn carries the heat to the coils located in the various air handling units that distribute heat to the different building areas. In the cooling season the refrigerant cycle is reversed.

The building owner utilizes 2400 volt primary electric service which, in addition to the heat pump load, supplies about 70 kw of base load, consisting mostly of lighting. The customer-owned transformer bank provides 440 volt secondary service for the heat pump and 120 volt supply for lighting and miscellaneous uses. Careful analyses show the estimated cost of electricity for heat pump operation. These estimates, for a normal degree-day basis of 4523 for Philadelphia, are given in one column

TABLE II — SUMMARY OF OPERATION DATA

METERED RESULTS — OCTOBER 1956 — SEPTEMBER 1957 DEGREE DAYS — 4190

Month	Base		Compressors Heating		Compressors Cooling		Auxiliaries Heating		Auxiliaries Cooling		Total Heat Pump		Grand Total[a]	
	Kw	Kwhr	Kw	Kwhr	Kw	Kwhr	Kw	Kwhr	Kw	Kwhr	Kw	Kwhr	Kw	Kwhr
1956 Oct.	71.2	19.090	36.8	1,440	49.6	1,710	22.4	4,250	22.4	2,300	68.0	9,700	138.0	29,780
" Nov.	71.4	17.420	84.8	6,460	41.6	510	21.6	7,810	19.2	730	101.6	15,510	172.0	33,920
" Dec.	71.4	15.800	84.6	7,560	—	—	20.0	8,490	—	—	100.8	16,050	167.0	32,850
1957 Jan.	71.4	19.680	80.0	15,760	—	—	24.0	12,450	—	—	98.4	28,120	167.0	48,790
" Feb.	71.5	16.790	57.6	9,380	—	—	24.4	9,800	—	—	80.0	19,180	146.0	36,970
" Mar.	70.0	17,390	56.0	12,930	35.2	170	23.2	10,950	17.6	230	72.0	24,280	142.0	42,670
" Apr.	70.0	16.590	48.0	6,290	35.2	960	21.2	7,000	17.2	1,030	64.8	15,280	134.0	32,870
" May	70.8	17.120	44.0	1,140	52.0	4,530	23.6	1,660	18.0	3,520	68.0	10,850	136.0	28,970
" June	68.4	15.560	—	—	47.0	3,740	—	—	20.8	4,660	69.8	8,400	138.0	24,960
" July	68.0	16.310	—	—	48.0	5,160	—	—	19.2	4,070	66.4	9,230	132.0	26,540
" Aug.	66.8	16.850	—	—	48.0	7,060	—	—	18.0	4,690	66.0	11,750	132.0	29,610
" Sept.	67.2	20.460	—	—	46.4	7,780	—	—	18.4	5,510	63.2	13,290	130.0	34,770
" 52nd Wk	69.0	4,370	—	—	29.6	760	—	—	23.2	1,680	47.7	2,440	116.0	7,060
Total		213,430		60,870		32,380		62,410		28,420		184,080		409,760

[a] Demand and energy (P. E. Co. primary service) include transformer losses.
Cost of Electric Energy on P. E. Co. Rate PD, Primary Distribution Power, including Fuel Adjustment of $0.00233/Kwhr, are Total Service — $7,489.62 ; Base Load Usage — $4,083.12 ; Heat Pump Operation — $3,406.50.

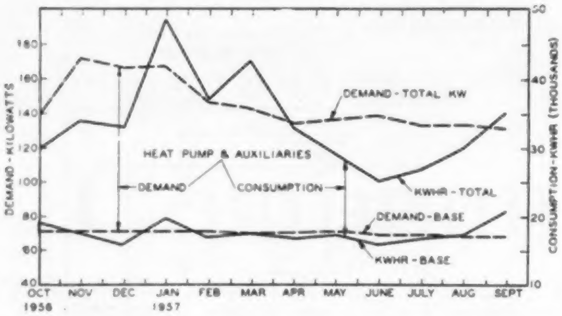


Fig. 2 Operating test data with day temperature setting at 74 F, and night temperature lowered to 65 F, with high demands on cold mornings

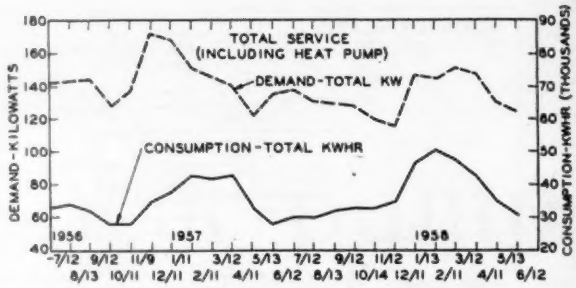


Fig. 3 Operating test data with temperature maintained steady throughout the whole night and day, show reductions of high demands on cold mornings

and compared with the actual costs established by test in the second column of Table I.

Calculations to determine the heating and cooling loads, worked out in collaboration with the manufacturer of the air source compound heat pump, are: net heat loss, 412,100 Btu per hr for a design condition of 0 F outside and 70 F inside; actual calculated load for cooling, 53 tons for an inside temperature of 80 F and outside temperature of 95 F (dry-bulb), 78 F (wet-bulb); capacity installed for cooling actually amounts to 60 tons and this could probably provide 77 F inside temperature during the summer.

It is apparent from Table I that the actual performance cost of the heat pump runs appreciably below the estimated cost. It is also important to note that during the test period, October 1956, through September 1957, the actual-degree days were 4190, below the normal of 4523 for Philadelphia.

Monthly metered results for the various elements of the load in the building are contained in Table II. The total cost of heat pump operation, for both heating and cooling, amounted to $3,406.50. If the customer had installed a conventional cooling system, the estimated annual cost of cooling would have been $1,700.18 and the balance of the $3,406.50, or $1,706.32, would represent the cost of heating (for the 4190 degree-days of the test period). Another separate analysis, Table III, gives a monthly breakdown of the combined heating and cooling costs for the test period.

During the first winter months of the test period the day setting of 74 F was changed to a night set-back temperature of 65 F. This had the effect of establishing high-demand on cold mornings.

As graphic test records and temperature data were obtained, a number of conferences were arranged with the personnel of the building owner's organization and the manufacturer of the heat pump. Ways of improving operations were discussed and the night set-back was discontinued. The test results began to reflect somewhat lower kw demand and, during the first year of operation, the practice of

TABLE III

MONTHLY BREAKDOWN HEATING AND COOLING COSTS COMBINED

Month		Total[a]
1956	October	$ 219.89
"	November	334.39
"	December	329.23
1957	January	429.69
"	February	312.40
"	March	350.89
"	April	260.42
"	May	225.46
"	June	214.81
"	July	210.01
"	August	233.04
"	September	242.52
"	52nd Week	43.75
	TOTAL	$3,406.50

[a] For 4190 degree-days

TABLE IV

ELECTRIC BILLING INFORMATION TOTAL LOAD — INCLUDING FUEL ADJUSTMENT

Period	Demand, Kw	Consumption, Kwhr	Cost[a]
7/12/56	142	33,100	$ 599.99
8/13	143	33,700	606.88
9/12	144	31,600	591.66
10/11	128	27,800	529.39
11/9	137	27,700	546.98
12/11	172	35,200	676.79
1/11/57	168	38,100	692.79
2/11	151	43,200	706.81
3/12	145	41,700	682.61
4/11	140	43,200	683.80
5/13	122	32,600	565.35
6/12	135	28,100	552.75
Total	1,727.0	416,000	$ 7,435.80
Av. Rate[a]	—	—	$0.0179
7/12/57	138	30,300	$ 577.15
8/13	131	30,400	568.30
9/12	130	31,500	575.77
10/14	128	33,100	585.57
11/12	120	32,800	568.91
12/11	116	35,400	591.29
1/13/58	147	46,500	736.40
2/11	145	50,500	767.62
3/12	152	48,000	765.40
4/11	148	42,800	715.26
5/13	131	35,100	616.85
6/12	125	30,900	560.17
Total	1,611.0	447,300	$ 7,628.69
Av. Rate[b]	—	—	$0.0171

[a] Degree days: 4239. [b] Degree days: 4603

night set-back was followed through March. In the second year, this was discontinued and it is apparent from the curve that the demand has decreased.

Table IV contains actual billing information for the total electric service to the building owner for a two-year period: one from July 12, 1956, to July 12, 1957, and the second from July 12, 1957, to July 12, 1958. The actual degree-days for these 2 periods were 4239 and 4603, respectively. Note the decrease in electric service demand, particularly in January, February and March of the second period, as compared with the first period. The increase in kwhr consumption for the second period reflects the effect of an increase in degree-days, as well as the elimination of the night set-back.

The overall results obtained with this heat pump installation have been quite satisfactory from a comfort, performance, and cost standpoint. The building location, construction and occupancy were well suited for heat pump operation. Building heat losses were minimal, and the lighting load and people load contributed favorably toward the heating cycle. These conditions were advantageous for a heat pump installation, and should be carefully considered in other types of buildings for which heat pumps are contemplated. An individual analysis, therefore, should be made to establish the relative economics for heating and cooling. This involves establishing the base electrical load, in demand and consumption, and superimposing the estimated increment demand and consumption for the heat pump compressor and auxiliaries on a month-by-month basis.

ACKNOWLEDGMENT

The author acknowledges the excellent cooperation obtained from The Ballinger Co. personnel, the York Corp. representatives, and Philadelphia Electric Company's Testing Div personnel, all of whom had a specific part in obtaining the test data and their compilation for this paper.

Acrylic resin insulation

FRED J. EMIG

ORVILLE J. SPAWN

Some recent developments in electrical insulation are of great importance to those who specify motors for the powering of household or commercial freezers, refrigerators and air conditioners.

The insulation on enameled magnet wire and the so-called space insulation used in winding motors and coils is extremely thin, but important. The insulating varnishes used to bond, support, and protect the windings of a motor are critical, too.

Introduction of Refrigerant-22 resulted in a requirement for more chemically inert and thermally stable electrical insulation so that optimum operating efficiency of hermetic units could be obtained. Because older enameled magnet wire and rag paper space insulation, even polyester and polyurethane insulations, did not meet all of the requirements for higher operating temperatures in hermetic units, acrylic resin insulating materials and wire enameled with polyester film space insulation are being used increasingly.

Fred J. Emig is Manager of Insulation Sales, Orville J. Spawn is Technical Supervisor, of the Finishes Div, E. I. duPont de Nemours & Co. This paper was presented at the Domestic Refrigerator Engineering Conference, ASRE 45th Semiannual Meeting, New Orleans, La., December 1-3, 1958.

Indeed, acrylic resin electrical insulation is the result of a careful study of the requirements of the enameled magnet wire and electrical manufacturing industries, followed by a specific research program to develop the products which these surveys indicated were needed.

Preliminary studies indicated that a water borne acrylic resin had many of the properties required in an improved insulating material. When compared with conventional insulations the acrylic resin was found to possess sufficiently improved thermal stability to allow use of Class B (130 C) temperature; increased solvent resistance, especially resistant to Refrigerant-22; equal or better physical and electrical properties; substantially no fuming during application; and potential low cost on volume production.

Subsequent development work led to the introduction of a line of acrylic resins, a family of copolymer latices. Various members of this polymer family are used to formulate a variety of insulating materials, such as magnet wire enamels, coatings for sheet insulation, laminating resins and coil or motor insulating varnishes.

Coated wire—Data presented herewith relate to 18 gauge copper wire, coated to nominal heavy build unless otherwise stated. Other sizes of wire were evaluated and results were found to be comparable. The results reported for the otherwise insulated magnet wire

control are averages obtained from several brands purchased on the open market.

Table I shows that magnet wire coated with acrylic resins has sufficiently improved flexibility, adhesion and scrape abrasion to assure good performance on the production line. This has been confirmed by production runs on automatic winding machines, in which the wire has been wound a little faster than present magnet wires. The acrylic does not solvent craze unless stretched more than 3-5%. Thus, if the production line can be adjusted to maintain wire stretch at 3 % or under, the subsequent annealing step before varnishing may be eliminated. The improved wet insulation resistance properties indicate better performance in the presence of moisture, which, of course, should not be present in a hermetic unit but may get into larger units during field servicing.

The initial properties of magnet wire coated with acrylic resin wire enamel at Class B temperature (130 C) are considered to be satisfactory. The dielectric strength at 130 C is 2000 volt per mil, compared with 2000 volt per mil for other enamels (R) at 105 C. The heat shock resistance is good. Cut through temperature is 300 C, compared with 240 C for others.

As shown in Fig. 1 the acrylic insulation has a lower initial resistance than does material "R." On heat aging, the former improves at 600 hr and over. The insulation resistance at 130 C is considered adequate for all practical applications. At Class B temperature (130 C) it is just as high in initial insulation resistance as nylon at Class A temperature (105 C); the latter has been used for years as Class A insulation.

Thermal Stability—This has been determined by the rate of loss of flexibility, coating weight and dielectric strength of varnished and unvarnished twisted wires, over a range of temperatures.

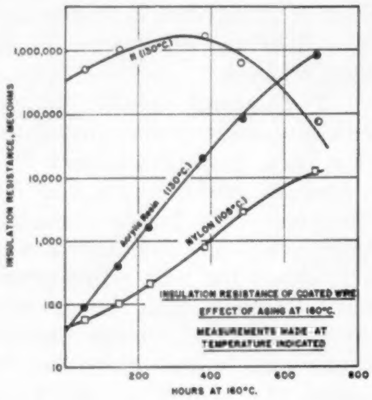


Fig. 1 Insulation resistance of coated wire

Weight loss was determined by periodically weighing a 6 ft length of loosely coiled wire. Weight loss data are given for 175 and 200 C in Fig. 2. These data show that at least a 30 C higher temperature is required for acrylic resin wire enamel to lose the same amount of film weight as did the compared material (R) in the same time.

Dielectric strength of unvarnished sample wire twists as a function of time and temperature are given on Fig. 3. These data show that the dielectric strength retention at 130 C is equivalent to that of type R at 105 C. Twists coated with certain varnishes show a 5-10 C improvement over unvarnished twists in the operating range. The best varnish for this purpose has not yet been established, although acrylic resin varnishes appear satisfactory. Since the acrylic is a Class A insulation having a maximum rating of 105 C, and thermal stability tests show that it is 25-30 C better, it should operate at a maximum temperature of 130 C or Class B temperature.

Solvent Resistance—The acrylic is unaffected by the solvents normally used to test magnet wire and by the varnish solvents encountered in coil impregnation. Acrylic resin magnet wire insulation has excellent resistance to Refrigerant-22. This has been tested by three methods: blistering, softening, and extractibles. Several blistering tests have been used. In the first, straight lengths of coated wire are immersed in Refrigerant-22 for 24 hr at room temperature in a suitable bomb. The Refrigerant-22 is then cooled to −40 C to prevent rapid evaporation of the refrigerant when the bomb is opened. The wires are removed, held for one min at room temperature, and baked 5 min at 140 C. The second is a modification in which the Refrigerant-22 is held at 115 F instead of room temperature for 24 hr. The third method involves heating the wire for 60 days at 250 F in Refrigerant-22 containing refrigerator oil and other components of the refrigerator motor. In all of these tests the acrylic showed excellent resistance to blistering and virtually no softening. In addition, it was found to cause less decomposition of the refrigerant than any other insulation tested. Fig. 4 compares wires enameled with the compared insulations after both were in the same blister test using Refrigerant-22. One is softened and blistered. The acrylic is neither softened nor blistered under the same conditions.

Insolubility is shown by the standard extraction test, using Refrigerant-22 as a solvent. Refrigerant-22 extracts 0.5% or less of solid

TABLE I

SOME INITIAL MAGNET WIRE PROPERTIES

	Acrylic Resin	Compared Resin
Flexibility-Adhesion Twist Test (Twists)	75	55
Scrape-Abrasion (Scrapes)	35	30
Resistance to Crazing	OK	Crazes
Dielectric Strength-25 C (Volt/mil)	2500	2500
Dielectric Strength (Volt/mil)	2000 (130 C)	2000 (105 C)
Wet Insulation Resistance (Days)	30-60	7-14
Resistance to Heat Shock	OK	Cracks
Cut Through Temperature (C)	300 C	240 C

TABLE II

TYPICAL 14 MIL ACRYLIC RESIN COATED GLASS FABRIC

Tensile Strength	— 250 × 250 lb/in. (WxF)
Crease Tensile Strength*	— 160 × 135 lb/in. (WxF)
Dielectric Strength**	— >600 volt/mil
Dielectric Strength** after 1 week at 98% R.H.	— >500 volt/mil
Dielectric Constant 1000 cps. 25 C	— 4.4
Dissipation Factor 1000 cps. 25 C	— 0.035
Toluene/Methanol Extractibles	— 0.7%
Refrigerant-22 Extractibles	— 0.2-0.3%
Insulation Resistance	— 10^{13}ohm/cm

* Creased with 10 lb. roller.
** ASTM D-295-55T short time test 0.25 in. electrodes.

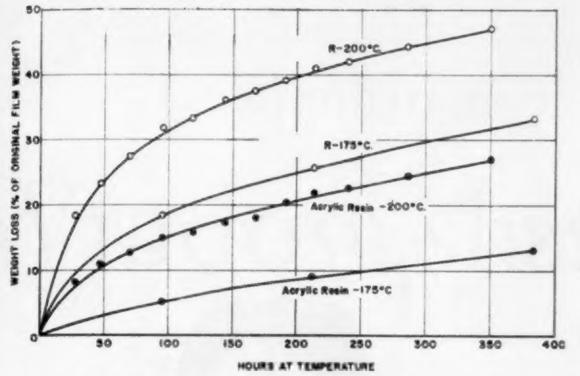
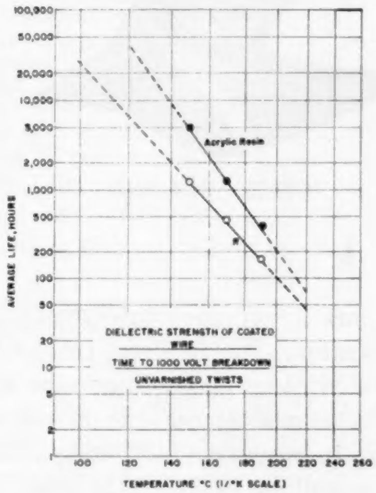


Fig. 2 Weight loss of coated wire

Fig. 3 Dielectric strength of coated wire



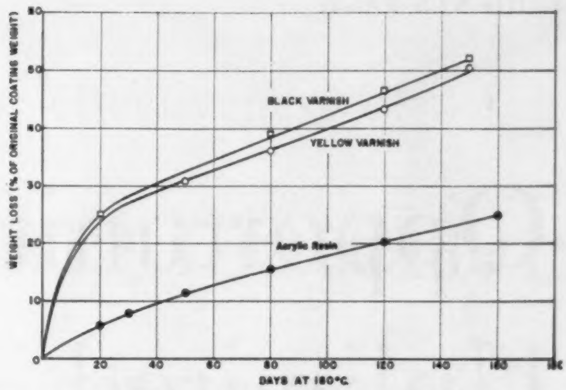
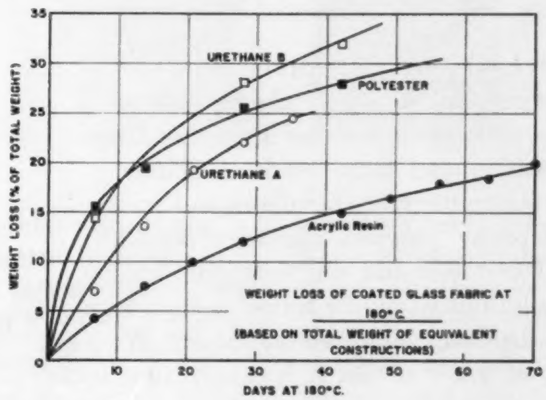


Fig. 4 Weight loss of coatings on glass fabrics at 60 C

Fig. 5 Weight loss of coated glass fibers at 180 C



material out of the acrylic compared with 2.5 to 6% for the other insulation.

Results from tests with Refrigerant-22 indicated that acrylic resin wire enamel is of outstanding value as insulation in hermetic refrigerator motors designed for normal or higher operating temperatures. Production tests are confirming these results.

Insulating varnishes – In order to make a complete insulating system possible, insulating varnishes of acrylic resin have been formulated and are currently undergoing extensive field tests. Results have been satisfactory to date. These are the original water dispersed acrylic resin varnishes that fit so well into the hermetic picture, while affording greater safety in plant operation.

Coated glass fabrics – Acrylic resin coated fabrics have been developed for use as sheet insulation in Class B and hermetic motor applications. The polymer which is used to coat the glass fabric is more flexible than that used in wire enamel and exhibits superior retention of flexibility on aging in air at elevated temperatures with only a slight sacrifice in the excellent solvent resistance of the wire enamel.

These products are available in several forms. Continuous rolls of single ply material can be supplied in thicknesses of 3-14 mils. Continuous seamed bias constructions can be supplied as well as straight cut products. Continuous laminates are available in thicknesses up to 30 mils, and thicker laminated constructions can be made in sheet form.

Several acrylic resin glass fabric products have been developed which have been fabricated successfully on automatic equipment into slot liners for fractional horsepower motors. These slot liners have withstood automatic winding, and the insertion of preformed coils. In larger motors ranging from 5 to 300 hp, laminates have been used for wedges and for slot liners, and bias-cut tapes have been used to wrap form wound coils. These products are also being evaluated for various transformer applications.

The properties of a 14 mil single ply construction are given in Table II. Of particular interest is the balance of high mechanical strength, good dielectric strength and low extractible content in toluene-methanol and Refrigerant-22. This excellent resistance to fluorinated refrigerants makes these products of particular interest for hermetic motor application. In addition to low extractible content, the coated fabrics do not blister. They remain tough on exposure to any of the fluorinated refrigerants, including Refrigerant-22.

(Continued on page 102)

Opportunities Unlimited

CECIL BOLING

In many respects, I regret that my term as President of ASHRAE is so near its close.

This has been a vastly stimulating year – the merger and its many closely related problems – and yet it is clear that through the cooperation and team work of officers, directors, committeemen, members and staff most astonishing progress can be reported. For one, I certainly did not anticipate that so many solutions would be found so soon to so many baffling inter-relationships. We should all be grateful that it is so.

Again, this has been for me a most personally gratifying year. The opportunity to attend so many Section, Chapter and Regional Meetings, to renew and extend friendships and to discuss mutual problems has been a highly rewarding one.

You have read in the JOURNAL of both newly established programs, indecisively considered ones and of things to come. Yet, there is much still to be accomplished, of course. Just how soon some of these things-to-come will be matters of reality is beyond projection at the moment, but that same cooperative, forward-looking spirit which has activated the new ASHRAE will be productive in the fullest sense in the time ahead.

I am looking forward to seeing many of you at Lake Placid this month. This should prove to be an extraordinary meeting from the standpoints of program, attendance and attractive location. The Program Committee has coordinated older commitments by each of the predecessor societies with newer and timely papers to produce a full but significant schedule. It will be worth your time and attention. As the first meeting of the new ASHRAE it offers an occasion of unusual depth.

Much has been said of the unity of objectives, purposes, people and opportunities that lie before ASHRAE. I doubt that enough has really been developed as yet to clearly define the great future that lies before us in the fields of heating, refrigerating, air conditioning and ventilating. It will only be with experience upon specific projects by our own Society, upon cooperative work with other societies and associations and in the fuller understanding of our mutual objectives and efforts that these things will become defined.

Let us look forward to and work together toward that bright tomorrow and rejoice in having had a part in its shaping.

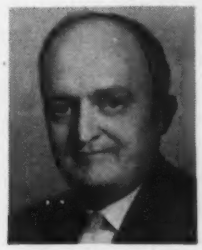
Determination of the effectiveness of

Window shading materials

on the reduction of solar radiation heat gain

RICHARD C. JORDAN
Member ASHRAE

JAMES L. THRELKELD
Member ASHRAE



When solar radiation is transmitted through a glass window and a shading material is interposed between the room and the glass, a portion of this energy is transmitted through the shade directly, unless opaque, a part is reflected from the shade back to the glass and a part is absorbed by the shade itself. That portion of the energy absorbed by the shading material warms the shade to a temperature above that of the surrounding air. This, in turn, results in the transmission of heat from the shade, both to the interior of the room and in most cases back to and through the glass. Thus, a portion of the radiant energy incident upon the shade is added to the room load and a portion is retransmitted through the glass to the outside. These researches are concerned with the development of methods for the determination of the effectiveness of shading materials in reducing solar loads transmitted through and from shading materials into room interiors and the comparison of typical experimental results obtained by two specific test methods.

Two different procedures were used. First, the University of Minnesota solar calorimeters were adapted to testing shading materials and the proportion of the incident solar energy passing through the shading materials located in the calorimeters was measured. Second, determinations were made of the transmissivities and reflectivities of the shading materials independent of the calorimeter tests and equations developed by which such physical factors could be used to calculate the shade transmission factors. Both procedures are described and typical test results compared.

Professor Richard C. Jordan is Head, Department of Mechanical Engineering, University of Minnesota. Professor James L. Threlkeld is with the Department of Mechanical Engineering, University of Minnesota. This paper was presented at the ASHRAE annual meeting, Lake Placid, N. Y., June 22-24, 1959.

SOLAR CALORIMETER TESTS

The two solar calorimeters used in these studies have been described in detail in an earlier publication.[1] These solar collectors or calorimeters are identical in construction and each is composed of a wooden enclosure, supporting frame, glass cover, and a copper collector plate with attached tubing for the absorption of heat loads. The calorimeter boxes are 94½ in. long, 48½ in. high, and 18 in. deep and are insulated with four in. of cork insulation on all sides excepting that containing the glass cover. The glass front is mounted on a steel frame and the glass opening itself is 39 in. x 85 in. The glass is ¼-in. water white plate glass with a total solar transmittance of 91% for normal incident solar radiation. The copper collecting plates are located 4½ in. behind the glass covers and are coated with a black absorbing surface having a spectral reflectance in the visible range of 0.0395. Heat passing through the glass covers and through any interposed shading materials is collected on the copper plates and hence carried away by the transfer fluid circulating in serpentine copper tubing soldered to the front side of the plates. The shading material under test was installed with the edges sealed between the glass cover and the copper plate at a distance of 1⅝ in. from the glass cover. Although the collectors are hinged to permit tilting to any position between horizontal and vertical, all tests were conducted with the collectors in the south-facing vertical position in order to simulate windows.

The collectors incorporated two separate fluid circulating systems, one for each collector. Each system consists of a storage tank for the circulating fluid, a pump, a refrigeration unit for the absorption of the collected heat, a flow controller, and a flow-rate sensing element and recorder. The fluid temperature in the storage tank was maintained at a constant value by a thermostatically controlled refrigeration system. In addition the flow controller maintained a constant outlet temperature by ac-

[1] Jordan, R. C. and Threlkeld, J. L., "Laboratory for Solar Energy Study at Minnesota," Heating, Piping, and Air Conditioning, May 1956, Pages 143-147.

tuating a valve which varied the flow rate so as to maintain the outlet temperature at a fixed point. The flow-rate sensing element and recorder provided a continuous record of the flow rate throughout the tests. Since the amount of heat transmitted through the shading materials was small, the flow rate was also determined by actual weight measurement.

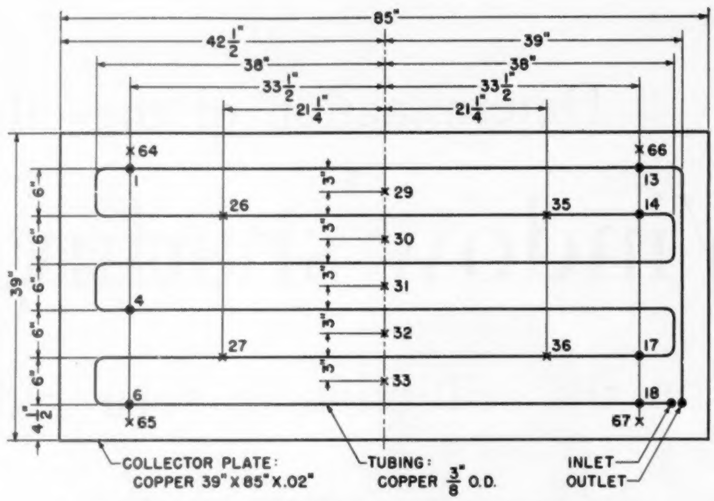
All temperatures were determined by copper-constantan thermocouples with the millivolt output measured by a self-balancing precision potentiometer. Four immersion thermocouples were used for measuring the inlet and outlet fluid temperatures of the calorimeters and twenty thermocouples for measuring the fluid and plate temperatures at various locations. In addition, 12 thermocouples were used for measuring the temperature of the shading material itself and 16 thermocouples for measuring the air temperatures inside the collector box. The locations of these thermocouples are shown in Figs. 1 and 2.

The incident total solar radiation was measured by means of a 50-junction Eppley pyrheliometer mounted in the vertical position. Diffuse radiation measurements were made by shielding the element of the pyrheliometer with a shading ball.

Determination of shade transmission factors from physical constants —The experimental shade transmission factor, E'_s, determined by the calorimeter tests is the ratio of the solar heat collected to the incident solar radiation, corrected for the reduction in incident energy as it passes through the cover glass. However, it is shown in Appendix A (Equation A-15) that the calculated shade transmission factor, E_s, for shading materials installed with edges sealed in such flat-plate calorimeters may be determined by the equation

$$E_s = \frac{1}{(1-\rho_s\,\rho_G)}\left[\frac{\alpha_p\,\tau_s}{(1-\rho_p\,\rho_{s,B})}+\frac{\alpha_s}{1+\frac{U_{so}}{C_{sp}}}\right] \qquad (1)$$

and, in the case of opaque shades, this reduces to



● IMMERSION THERMOCOUPLES FOR MEASURING FLUID TEMPERATURES.
× THERMOCOUPLES ON BACK SIDE OF PLATE FOR MEASURING PLATE TEMPERATURES.

Fig. 1 Location of plate and fluid temperature measuring thermocouples

$$E_s = \frac{\alpha_s}{(1-\rho_s\,\rho_G)\left(1+\frac{U_{so}}{C_{sp}}\right)} \qquad (2)$$

In order to evaluate this equation, it is necessary to determine experimentally the transmissivities and reflectivities of the shading materials, the absorptivity of the blackened copper plate and the reflectivity of the calorimeter cover glass.

Reflectivity measurements with normally incident radiation were made by means of a hemispherical integrating radiometer.[2] This radiometer was equipped with a thermopile and a test surface located at conjugate foci on the diameter of the hemisphere. The solar radiation passing through the aperture of the radiometer was first allowed to impinge on the thermopile as an indication of the incident solar radiation. Next, solar energy was reflected from the test surface onto the thermopile with the necessary integration of energy reflected at all angles accomplished by the hemispherical surface.

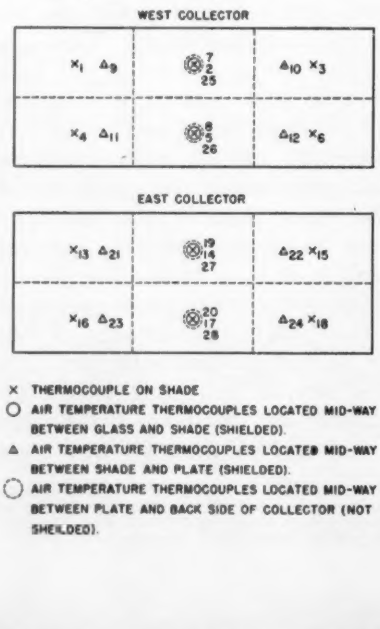
The ratio of the energy reflected to the thermopile from the test surface to that directly incident onto the thermopile provides a measure of the relative reflectivity of the test surface when compared to a roughened surface with known absolute reflectivity. The reference used was a magnesium oxide surface for which the reflectivity in the solar spectrum is approximately 0.95.

Additional reflectivity measurement checks were made with an integrating sphere radiometer in which the inside of the sphere was coated with magnesium oxide so that radiation reflected from the surface under test provided a uniform radiation field throughout the sphere and measurement of this and comparison with a surface of known reflectivity provides a measurement of the relative reflectivity.

Transmissivity measurements of the cover glass used on the calorimeters, the shading materials, and combinations of shading materials and glass were measured by

[2] Birkebak, Richard C. and Hartnett, James P., "Measurements of the Total Absorptivity for Solar Radiation of Several Engineering Materials." Transactions of the American Society of Mechanical Engineers, 80:373-78. February, 1958.

Fig. 2 Location of solar collector air temperature measuring thermocouples



× THERMOCOUPLE ON SHADE
○ AIR TEMPERATURE THERMOCOUPLES LOCATED MID-WAY BETWEEN GLASS AND SHADE (SHIELDED).
△ AIR TEMPERATURE THERMOCOUPLES LOCATED MID-WAY BETWEEN SHADE AND PLATE (SHIELDED).
◯ AIR TEMPERATURE THERMOCOUPLES LOCATED MID-WAY BETWEEN PLATE AND BACK SIDE OF COLLECTOR (NOT SHEILDED).

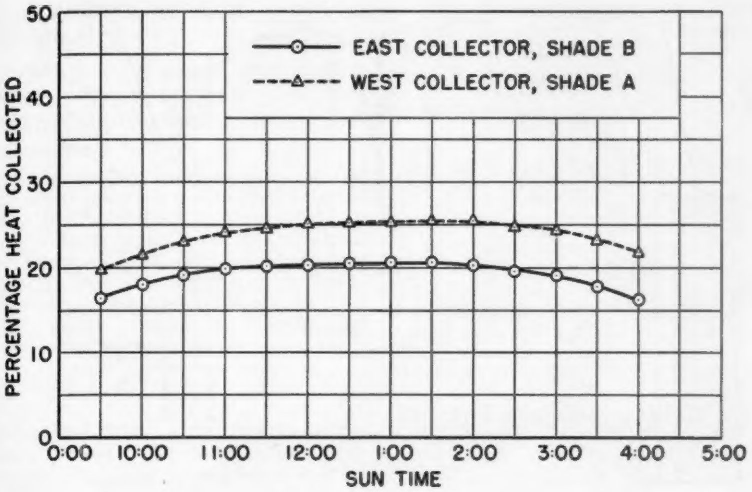


Fig. 3 Typical solar calorimeter test results comparing two shading materials

means of a 9 x 18 x 18-in. chamber placed over a 50-junction Eppley pyrheliometer. The front of this chamber was open to permit the insertion of a section of glass similar to that used on the collectors and panels of the shading materials. This permitted measurements of the incident solar radiation without the interposed glass or shading material and of the incident solar radiation transmitted through the shade and glass. Determinations of the absorptivity factors, α_s, were made by the equation $\alpha_s = 1 - \tau_s - \rho_s$, once the transmissivities and reflectivities were known.

Test results – In these studies five different types of shading materials were used and each may be briefly described as follows:

Shade Material A
Bleached cotton cloth, vinyl resin solution coating, and fireproofed, partially translucent, slightly glossy, buff-colored surfaces.
Shade Material B
Bleached cotton cloth, vinyl resin solution coating, partially translucent, white in color, low surface gloss.
Shade Material C
Bleached Kraft paper base, vinyl resin solution coating, partially translucent, embossed white surfaces on both sides.
Shade Material D
Cloth base, treated and coated with vinyl resin, opaque, slightly glossy white finish on both sides
Shade Material E
of the material.

Vinyl-coated cotton base, opaque, laminated with embossed aluminum foil on one side and white, coated surface on the other.

It was found from reflectivity measurements that the two sides of materials A, C, and D had different reflectivities. These materials were fitted in the solar calorimeter with the dull side (the side with lower reflectivity) facing out, according to the manufacturer's instructions. Material E was tested with the embossed aluminum foil facing out. These five shading materials were tested in the solar calorimeters by sealing the material under the back of the glass cover with an air space of 1⅝ in. between the cover and the shade.

This provided an idealized condition in which no air movement could occur around the edges of the shading material and differs from practice unless channels or similar devices are provided in which the shade may be raised or lowered. The testing and evaluation are comparable to that for

TABLE I

Comparison of Experimental and Calculated Shade Transmission Factors for Solar Calorimeter (Shade edges sealed)

Shade Material	Experimental Shade Transmission Factor E'_s	Calculated Shade Transmission Factor E_s
A	0.28	0.30
B	0.23	0.26
C	0.23	0.26
D	0.18	0.18
E	0.14	0.14

shading materials in a fully drawn position with no attempt to simulate partially drawn shades.

During actual testing, one calorimeter was equipped with shading material B as a standard for comparison since this material appeared to have a comparatively high transmission factor and therefore allowed better accuracy in determining heat collected. The alternate shading material was installed in the second collector. An additional test was made with shading material D compared directly with shading material E since these two both appeared to have quite low shade transmission factors under identical solar conditions.

Fig. 3 shows typical curves obtained for the percentage of incident solar heat collected on a cloudless day by the calorimeters when equipped with different shading materials. The percentage of heat collected was comparatively constant through the hours near solar noon but dropped rapidly during the late hours and rose rapidly during the earlier morning hours when the angles of incidence between the sun and the calorimeters were comparatively high.

Since the experimental shade transmission factor, E'_s, as determined by the calorimeter test is inherently a measurement made under controlled conditions similar to actual use and does not necessitate the determination of the physical constants required for evaluation of E_s by Equation (1), the calorimeter test procedure was regarded as yielding the more accurate results. Both the reflectivity and transmissivity apparatus and measurement procedures require further refinement before equivalent confidence can be placed in the calculated shade transmission factor, since the equation to determine E_s is quite sensitive to small variations in these measurements.

Table I shows a comparison of the calorimeter test results and the calculated shade transmission factors for all shading materials tested. The sensitivity of the results obtained in determining the calculated shade transmission factors by use of Equation (1) may be illustrated by analyzing the calculations for shade E. The average of the experimentally determined values

of the reflectivity, ρ_s, was 0.85 and this value provided an exact check between the experimental and the calculated shade transmission factors, $E'_s = E_s = 0.14$. Yet the range of values for ρ_s from which this average was obtained permitted a range of values of E_s between 0.17 and 0.11.

Shade transmission factors for windows in rooms—Since configuration of the glass cover and tight fitting shade in the solar calorimeter absorbing space is not identical to that of a window and a tight fitting shade in the usual room it can be shown (see Appendix B) that instead of Equation (1) the solar transmission shading factor for a window equipped with a tight fitting shade is given by

$$\varepsilon_s = \frac{1}{(1-\rho_s\,\rho_G)}\left[\tau_s + \frac{\alpha_s}{1+\dfrac{U_{so}}{f_i}}\right] \qquad (3)$$

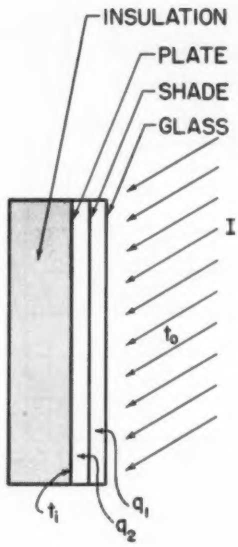
It is also shown (Appendix B) that for a non-opaque shade the relationship between ε_s and E_s or E'_s is given by the equation

$$\varepsilon_s = E'_s\left[\frac{\tau_s + \dfrac{\alpha_s}{1+\dfrac{U_{so}}{f_i}}}{\tau_s + \dfrac{\alpha_s}{1+\dfrac{U_{so}}{C_{sp}}}}\right] \qquad (4)$$

and for an opaque shade by the equation

$$\varepsilon_s = E'_s\left[\frac{1+U_{so}/C_{sp}}{1+U_{so}/f_i}\right] \qquad (5)$$

Using these relationships of Equations (4) and (5) the values of ε_s may be calculated. Table II shows a summary of calculations for the shade transmission factor ε_s for windows for angles of incidence between 0 and approximately 60 deg.



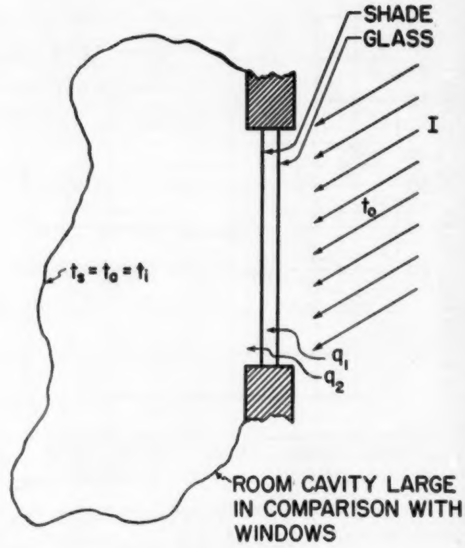


FIG. A-1

FIG. B-1

Fig. A-1 Schematic solar calorimeter with shade

Fig. B-1 Shaded window in relation to room

SUMMARY

Window shading material transmission factors for five different materials were determined both by solar calorimeter tests, and by calculation through independently measured reflectivities and transmissivities of the materials. Equations have been derived for calculation of the transmission factors, both in application to a calorimeter and also in application to a window and room. The checks between the experimental and calculated shade transmission factors are good in all cases.

However, the sensitivity of the calculated transmission factors to determination of the reflectivity and transmissivity is such that experimental variations in these values materially affect the overall transmission values. In those cases where the physical constants may be reliably evaluated the transmission factors may be calculated accurately, but in those cases where these values may be questioned, calorimeter tests appear necessary.

ACKNOWLEDGMENT

The authors acknowledge the cooperation of the Reynolds Metals Company, whose financial assistance made possible these experimental studies. The solar calorimeters were developed and the calorimeter analytical studies were made in conjunction with the solar energy research program sponsored at the University of Minnesota by the ASHAE now **American Society of Heating, Refrigerating and Air-Conditioning Engineers.**

Mr. J. F. O'Keefe, Product Development Dept., and Mr. A. J. Castle, Foil Development Laboratory, Reynolds Metals Company, provided valuable assistance in coordinating the experimental tests. Materials for the test program were provided by Mr. B. E. Beale, Director of Development, Joanna Western Mills.

The authors also acknowledge the able assistance of Messrs. Harrison Benjamin and Benjamin Liu, Research Assistants in the Department of Mechanical Engineering, in conducting the calorimeter tests, and of Mr. Richard Birkebak, Instructor in Mechanical Engineering, in making the reflectivity measurements.

TABLE II

Summary of Factors Involved in Calculation of Shade Transmission Factors for Windows Located in Rooms (Shade edges sealed)

Shade Material	E'_s	τ_s	α_s	U_{so}	U_{so}/C_{sp}	U_{so}/f_i	ε_s
A	0.28	0.135	0.250	0.86	0.716	0.589	.29
B	0.23	0.147	0.143	0.86	0.716	0.589	.24
C	0.23	0.125	0.185	0.86	0.716	0.589	.24
D	0.18	0.0	0.27	0.86	0.716	0.589	.19
E	0.14	0.0	0.155	0.32	0.267	0.219	.15

APPENDIX A

Determination of shade transmission factor for shaded collector plate with plate temperature equal to outdoor temperature

Fig. A-1 shows schematically a flat plate collector with a shading material interposed between the collector plate and the cover glass. In this analysis it is assumed that the collector plate is maintained at a temperature equal to the outdoor ambient temperature.

The solar radiation transmitted through the glass cover is

$$Q_1 = F_s A_s \tau_G I \qquad \text{(A-1)}$$

A part of this energy may be transmitted through the shade directly, a part reflected and a part absorbed by the shade. Part of the reflected energy is reflected back again to the shade by the glass cover and this tends to increase the radiation transmitted and absorbed. The total radiation directly transmitted through the shade is

$$Q_2 = \tau_s Q_1 (1 + \rho_s \rho_G + \rho_s^2 \rho_G^2 + \dots) = \frac{\tau_s Q_1}{1 - \rho_s \rho_G} \qquad \text{(A-2)}$$

The total radiation absorbed by the shade is

$$Q_3 = \frac{\alpha_s Q_1}{1 - \rho_s \rho_G} \qquad \text{(A-3)}$$

The radiation absorbed by the shade will cause an increase in shade temperature above the plate temperature. If the shade has negligible temperature drop and is at a uniform temperature, t_s, part of the energy absorbed will be transferred outwardly through the glass, or

$$Q_3 = Q_4 + Q_5 \qquad \text{(A-4)}$$

or

$$Q_3 = C_{sp} A_s (t_s - t_p) + U_{so} A_s (t_s - t_o) \qquad \text{(A-5)}$$

where

$$U_{so} = \frac{1}{\dfrac{1}{C_{sG}} + \dfrac{X_G}{k_G} + \dfrac{1}{f_o}} \qquad \text{(A-6)}$$

Since in the tests, the collector plate temperature was maintained equal to the outdoor air temperature, Equation (A-5) may be written as

$$Q_3 = A_s (t_s - t_o)(C_{sp} + U_{so}) \qquad \text{(A-7)}$$

By Equations (A-1), (A-3), and (A-7)

$$t_s - t_o = \frac{\alpha_s F_s \tau_G I}{(1 - \rho_s \rho_G)(C_{sp} + U_{so})} \qquad \text{(A-8)}$$

By Equations (A-1), (A-3), (A-4), (A-5), and (A-8)

$$Q_4 = \frac{\alpha_s F_s \tau_G I}{1 - \rho_s \rho_G} \left[1 - \frac{U_{so}}{C_{sp} + U_{so}}\right] A_s \qquad \text{(A-9)}$$

The directly transmitted radiation which is absorbed by the collector plate is

$$Q_6 = \frac{\alpha_p}{(1 - \rho_p \rho_{s,B})} Q_2 \qquad \text{(A-10)}$$

and by Equations (A-1), (A-2), and (A-10)

$$Q_6 = \frac{\alpha_p \tau_s F_s \tau_G I A_s}{(1 - \rho_p \rho_{s,B})(1 - \rho_s \rho_G)} \qquad \text{(A-11)}$$

The total energy absorbed by the plate is

$$Q_A = Q_4 + Q_6 \qquad \text{(A-12)}$$

and by Equations (A-9), (A-11), and (A-12)

$$Q_A = \frac{F_s A_s \tau_G I}{(1 - \rho_s \rho_G)} \left[\frac{\alpha_p \tau_s}{(1 - \rho_p \rho_{s,B})} + \frac{\alpha_s}{1 + \dfrac{U_{so}}{C_{sp}}}\right] \qquad \text{(A-13)}$$

If the shade transmission factor E_s is defined as

$$E_s = \frac{Q_A}{Q_1} \qquad \text{(A-14)}$$

Then by Equations (A-1), (A-13), and (A-14)

$$E_s = \frac{1}{(1 - \rho_s \rho_G)} \left[\frac{\alpha_p \tau_s}{(1 - \rho_p \rho_{s,B})} + \frac{\alpha_s}{1 + \dfrac{U_{so}}{C_{sp}}}\right] \qquad \text{(A-15)}$$

APPENDIX B

Determination of shade transmission factor for a shaded window

Fig. B-1 shows schematically a shaded window in relation to a comparatively large room. By means analogous to those of Appendix A, we have for the total energy transferred to the room

$$Q_A = Q_2 + Q_4 \qquad \text{(B-1)}$$

where

$$Q_2 = \frac{\tau_s F_s A_s \tau_G I}{(1 - \rho_s \rho_G)} \qquad \text{(B-2)}$$

$$Q_4 = f_i A_s (t_s - t_i) \qquad \text{(B-3)}$$

and

$$Q_4 = \frac{\alpha_s F_s A_s \tau_G I}{(1 - \rho_s \rho_G)} - U_{so} A_s (t_s - t_o) \qquad \text{(B-4)}$$

By Equation (B-3) and (B-4)

$$t_s = \frac{\dfrac{\alpha_s F_s \tau_G I}{(1 - \rho_s \rho_G)} + f_i t_i + U_{so} t_o}{f_i + U_{so}} \qquad \text{(B-5)}$$

By Equations (B-3) and (B-5)

$$Q_4 = \frac{\alpha_s F_s A_s \tau_G I}{(1 - \rho_s \rho_G)\left(1 + \dfrac{U_{so}}{f_i}\right)} + U A_s (t_o - t_i) \qquad \text{(B-6)}$$

where

$$U = \frac{1}{\dfrac{1}{U_{so}} + \dfrac{1}{f_i}} \qquad \text{(B-7)}$$

Thus by Equations (B-1), (B-2), and (B-6)

$$Q_A = \frac{F_s A_s \tau_G I}{(1 - \rho_s \rho_G)} \left[\tau_s + \frac{\alpha_s}{1 + \dfrac{U_{so}}{f_i}}\right] + U A_s (t_o - t_i) \qquad \text{(B-8)}$$

If we define the shade transmission factor as

$$\epsilon_s = \frac{Q_A}{Q_1} = \frac{Q_A}{F_s A_s \tau_G I} \qquad \text{(B-9)}$$

Then by Equations (B-8) and (B-9)

$$\epsilon_s = \frac{1}{(1 - \rho_s \rho_G)} \left[\tau_s + \frac{\alpha_s}{1 + \dfrac{U_{so}}{f_i}}\right] + \frac{U (t_o - t_i)}{F_s \tau_G I} \qquad \text{(B-10)}$$

Equation (B-10) is general and expresses the performance of the shade both with respect to solar transmission and to normal heat transmission. For solar effects only, or with $t_o = t_i$, Equation (B-10) reduces to

$$\epsilon_s = \frac{1}{(1 - \rho_s \rho_G)} \left[\tau_s + \frac{\alpha_s}{1 + \dfrac{U_{so}}{f_i}}\right] \qquad \text{(B-11)}$$

In Equation (A-15) of Appendix A, the term $\alpha_p/(1 - \rho_p \rho_{s,B})$ was essentially unity for all shading materials studied. With this approximation we may write by Equations (A-15) and (B-11)

$$\epsilon_s = E_s \frac{\tau_s + \dfrac{\alpha_s}{1 + U_{so}/f_i}}{\tau_s + \dfrac{\alpha_s}{1 + U_{so}/C_{sp}}} \qquad \text{(B-12)}$$

NOMENCLATURE

A_s = area of shade, sq ft

C_{sG} = thermal conductance from outside surface of shading material to inside surface of cover glass or window, Btu per (hr) (sq ft) (F)

C_{sp} = thermal conductance from inside surface of shading material to collector plate, Btu per (hr) (sq ft) (F)

(*Continued on page* 101)

ABSTRACTS OF TECHNICAL AND CONFERENCE PAPERS

FIRST TECHNICAL SESSION—MONDAY, JUNE 22, 9:30 A.M.

Turbine-driven centrifugal refrigerating systems

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Member ASHRAE

Many factors enter into the analysis and determination of the type of refrigeration system to be used when steam is available either from existing or contemplated boiler capacity as a power source. Pointing out the advantages of using steam turbine-driven centrifugals, as compared with steam-operated absorption refrigeration systems, the author considers operation economy, maintenance cost, noise and vibration, installation cost, and first cost. The operation economy factor is broken down into eight sub-factors, including heat balance, cost of operators, ease of operation, etc. All of these factors intertwine to evolve the over-all cost of the system. Other features are also considered. For example, the author points out that today's high caliber rotative machinery is both statistically and dynamically balanced, virtually eliminating vibration. Thus, noise from the refrigeration machines becomes unimportant in the over-all picture.

Based on the information in the paper it is concluded that consulting engineers and owners should investigate all of the factors which influence the selection of a type of refrigeration system for their air conditioning needs. Also, it is suggested that a little extra time taken at the design stage may pay for itself many times over in the important consideration of operating cost which is a large part of the owning cost and, in fact, may at times justify a larger initial investment.

Effects of refrigerant properties on compressor dimensions

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Centrifugal compressors for refrigeration, especially for the chilling of large volumes of water for use in relatively large air conditioning systems of the central station type, have gained broad acceptance. The original concept has been developed and expanded, with a multitude of improvements, to the point where refrigeration machines using centrifugal compressors are big business. This development has paralleled and hinged upon the simultaneous development of other machine components, and particularly has been influenced by the availability of suitable refrigerants.

The heat exchangers used in association with a compressor to create a complete machine are the major components so far as bulk and weight are concerned, and show high in the over-all machine cost. Thus, the heat exchangers deserve as much attention from the designer as the compressor, and the design of all the components is obviously inter-related.

This discussion concentrates on the effect of refrigerant choice upon centrifugal stage design, with a discussion of the various factors, including refrigerants, which limit and influence the optimum design. To maintain an acceptable over-all machine performance, compressor performance deficiencies must be compensated by adding heat exchanger surface.

In their paper the authors present a review of the

major factors which affect peak design performance of any centrifugal compressor stage for any application with any refrigerant vapor.

In this technical paper, some typical peak performance charts are presented, supported by test data and other information in the reference material, which can be used to estimate the required impeller dimensions, speeds and relative peak efficiency levels for centrifugal stages operating with any refrigerant, at any load and "lift" requirement. Specific comparisons are presented for typical 300 Ton Centrifugal Air Conditioning Compressors (both single stage and two stage, with economizer), using Refrigerants 11, 12, 113, and 114.

Free piston compressor in an air conditioning system

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R. W. KING
Assistant Division Chief
Battelle Memorial Institute

Under the sponsorship of the American Gas Association, Battelle Memorial Institute has developed a small free-piston refrigerant compressor for residential air conditioning. This compressor is a single-piston unit which has a capacity of 3 ton at its design speed of 1500 cycles per minute. The fuel is injected into the power cylinder at one end to furnish the power to compress the refrigerant at the opposite end. When the unit is running, the piston reciprocates continuously between the power and compressor cylinders with its motion controlled solely by the gas forces of the various chambers. Starting is accomplished by positioning the piston at its lower limit of travel and admitting refrigerant from the evaporator to drive the piston up on its starting stroke.

A major component of the free-piston compressor is the reverse leakage seal which consists of a high-pressure oil gland that prevents the loss of refrigerant from the system. The oil gland is maintained by a self-contained circulation system while hydraulic sealing rings prevent the loss of the sealing oil.

Stable operation of the free-piston compressor is obtained over a wide range of evaporator and condenser saturation temperatures. The unit has a high over-all efficiency with over 80% of the heat input to the engine being delivered as cooling effect in the evaporator. This high efficiency results in an operating cost that is substantially lower than that of electric-motor-driven units.

The condensing unit, which is completely air cooled, is a compact package which measures 50 x 30 x 33 in. Due to the simple construction of the free-piston compressor, the initial cost of the condensing unit is competitive with electric-motor-driven units. These factors, coupled with the low operating cost, should result in the free-piston refrigerant compressor being a strong contender as a power source in the air conditioning market.

Automatic computer for fan testing

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With this apparatus complete blower performance determinations are made in 15 min instead of the approximate four hours normally required for test and calculations. Performance at any desired speed is plotted automatically over the entire operating span of a blower in terms of brake horsepower or watt input, static pressure and flow. Data are reproducible within 1%, while data obtained in conventional chambers are within 4%.

Pressure measurements are made both with manometers and pressure transducers. Brake horsepower is measured with a torqueometer and watts input to direct drive units are determined with a thermal-converter. Power and static pressures are converted to electrical equivalents and fed, independently, through appropriate "weighting" networks to the abscissa of a two pen function plotter.

Pressure differential across selected nozzle combinations is first converted to an electrical signal, and then presented to a square root taking potentiometer. The signal which is then proportional to $\sqrt{\Delta p}$, is sent through a nozzle "weighting" network and scale factor circuit into the function plotter and represents air flow. In this way pressure characteristics, flow and power consumption are graphed continuously as chamber resistance is varied from maximum to minimum. Where necessary, electrical signals are modified to compensate for temperature and pressure and thus all data obtained are automatically corrected to standard air density.

Experience to date indicates that with proper maintenance and careful selection of circuit parameters trouble-free and accurate performance can be expected from this equipment.

Evaluation of filters for removing irritants from polluted air

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N. A. RICHARDSON
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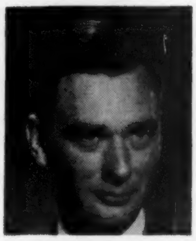
Department of Engineering
University of California

Two air-filter media were evaluated by their effectiveness in reducing human sensory irritation resulting from Los Angeles smog. The sensory response of one group of subjects working in a filtered atmosphere was compared with the response of another similar group working in a non-filtered atmosphere in identical, adjacent rooms. Sensory response was measured daily and simultaneous measurements of the physical composition of the air were obtained.

Much of the testing was with activated carbon filters varying in air detention time between 0.032 and 0.0030 sec. A significant decrease in irritation was recorded over the entire range of air detention times. Differences in effectiveness with respect to air detention time were not statistically significant, although a trend of decreasing effectiveness was observed as air detention time was reduced. Effectiveness of activated carbon in removing oxidants was directly related to detention time. NO_2 was reduced by activated carbon during its early use.

A particulate filter which effectively removes particles having a diameter less than 0.05 micron was also tested. No decrease in sensory irritation was detected. Correlations computed between measurements taken in the non-filtered atmosphere indicate that sensory irritation is highly related to oxidant level and moderately to temperature. Other data related to this study are reported also.

Analyses of chimney design and performance



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Royal Military College of Canada

Most methods commonly used in the design of chimneys are applicable only to a limited range of chimney heights, gas flows, etc. Some other methods rely on trial and error solutions which become long and unwieldly. This paper outlines a design procedure based on the theory as presented in a previous paper entitled, "A Fundamental Analysis of Chimney Performance." The chimney diameter, or cross-sectional area, which is obtained, results in optimum operation from an efficiency point of view. In order to select this chimney diameter it is necessary to know only the chimney height and the gas flow rate.

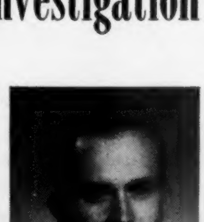
Once the chimney diameter has been selected it is possible to proceed through a number of steps to an actual draft that would be obtained. This draft is arrived at by taking into consideration the friction losses in the smoke pipe connection to the chimney, the losses in the vertical stack and the heat losses occurring in the vertical stack.

The entire method is not restricted to any particular fuel since the primary variable is the weight of flue gas flowing in lb per hr. Certain simplifying assumptions had to be made in order to keep the method reasonably simple. Some of these assumptions were: (1) Ambient outside air temperature of 60 F which would represent possibly the worst condition of operation, (2) surface roughness of the flue liner indicated by a relative roughness figure of 0.001. This would represent an average figure for normal liners. A revision of this figure would be required if a stainless steel flue were used such as the type in prefabricated chimneys. (3) Flue gas temperature at the chimney inlet was assumed to fall in the range 300 to 700 F. If a special case were encountered where the temperature was outside this range then a new graph could be plotted for the desired temperature.

A sample solution is included in the paper, which gives a clear step by step analysis of the sort of design that can now be handled making use of the theory and graphs as presented.

THIRD TECHNICAL SESSION—TUESDAY, JUNE 23, 9:00 A.M.

Investigation and control of refrigerator noise



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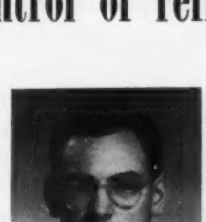

J. P. LAUGHLIN
Product Engineering Dept
Whirlpool Corporation

Test processes, devised to find solutions to some of the problems met in the control of refrigerator noise, were developed in the Evansville Div of the Whirlpool Corporation.

A sound test room was designed specifically for the test processes. The fundamentals of its design, and a description of the instruments used in it for measuring sound are discussed, together with an analysis of the methods of interpreting results, including the concepts of decibels, sones and the interrelationship between the two. Methods of testing and types of measurement are discussed in detail for compressors alone, units, and complete refrigerators, with special attention given the responsibility of both the quality control department and the engineering department in controlling noise. Descriptions are included of tests run in a special quality control, noise test cell for compressors, a labyrinth test for completed units, and finally the quality control sound room for the complete product.

Corrections for cabinet vibration, fan noise, and combinations of frequencies causing beats are given as practical examples of noise control in refrigerators. Also, some instances are shown where the judicious choice of designs and materials give desirable performance effects along with noise reduction.

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President

D. D. WILE
Second Vice President

W. A. GRANT
Third Vice President

R. H. TULL
Fourth Vice President

ASHRAE ANNUAL MEETING WILL BE

New officers of ASHRAE will assume their duties on Monday, June 22, at the Welcome Luncheon. Arthur J. Hess

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Program Chairman
Chairman, First Session

P. N. VINTHER
Program Vice-Chairman
Chairman, Second Session

W. E. FONTAINE
Chairman, Third Session

R. A. LINE
Chairman, Fourth Session

Investigating household refrigerator compressor noise

Discussing some methods for investigating certain machine noise problems, such as the noise from the small compressors of the type used on household refrigerators, the author takes the point of view of an engineer who desires to make a systematic study with quantitative measurements and a minimum of trial and error. Attention is given to major questions of measurement, the answers to which are not well established in reference material.

Noise studies involve two different phases, the physical or engineering phase and the human phase. What is noisy is determined by different criteria in the human phase rather than in the engineering phase. These criteria fall into two groups: one is loudness and the other annoyance. An irritating sound may be a low level sound with certain undesirable frequency characteristics. A household refrigerator noise may be a low level sound, and yet be considered noisy because the sound is annoying. Some tests indicate that disturbance is caused both by high frequency components, and by discreet frequency components.

The author describes a method for studying a compressor, and gives quantitative measurements indicating what should be done in order to quiet the compressor. A facility consisting of an echoless chamber, microphone, sound level meter, and spectrum analyzer is used to test a compressor with its normal charge and load, as these are the most desirable conditions. Other aspects of the problem, such as annoyance, masking, and possible tests, are also discussed.

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Purdue University

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HELD AT LAKE PLACID CLUB

will succeed for a six month term Cecil Boling, who was the first President of ASHRAE. Others move up.

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Chairman, Seventh Session

W. L. McGRATH
Chairman, Eighth Session

Acoustical testing of mufflers for refrigeration systems

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The performance of mufflers designed for use on the exhausts of internal combustion engines is fairly well documented. There is not such an extensive reference for mufflers used in refrigeration systems. This paper presents the results of one phase of a project to adapt acoustical methods of muffler design and

evaluation to mufflers which are to be used in the discharge line of rotary refrigeration compressors. The emphasis is on experimental methods of evaluation and problems encountered in correlating measured results in an actual system with predicted results in an ideal system.

The acoustical test method consists of measuring the transmission loss of the muffler by comparing the sound power incident upon the muffler with the power transmitted by it. A standing wave tube is the principal piece of experimental apparatus used in the determinations here reported.

The suppression of oscillations in gas-fired residential heating equipment

Oscillations, as differentiated from vibrations and other noise sources, have preoccupied manufacturers, installers and users of residential heating equipment for many years. Through experience gained by trial and error, a number of different suppression techniques have been found which are effective in some, but not all, circumstances. Lack of understanding of the causes of oscillations forestalled understanding and oil-fired residential heating equipment, analytic techniques are outlined in simplified form for ready reference. Since the mechanism of oscillation for each type of burner unit differs, the authors discuss both types of burner. Two techniques are outlined for the analysis of data, which will establish the location of regions of oscillation and non-oscillation in terms of easily measured furnace and burner variables. Using

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the reasons for the uncertainty of success of the suppression techniques. The entire oscillation problem has been further aggravated in recent years by the use of smaller, more compact heating units placed near the living quarters of residences. Also, people in general have become more noise conscious.

Discussing the fundamental causes and methods of suppressing oscillations and pulsations in both gas- this information as a basis, estimates can be made as to the vulnerability to oscillation of new heating designs with this same arrangement of burner at port level.

In connection with other units where oscillation problems exist, estimates can be made of the necessary changes required to produce an oscillation free unit.

FOURTH TECHNICAL SESSION—TUESDAY, JUNE 23, 9:00 A.M.

Solar radiant gains through directional glass exposure

R. D. CRAMER
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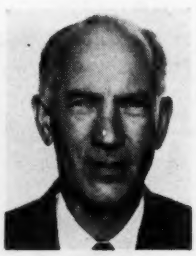
Dept Home Economics
University of California

L. W. NEUBAUER
Professor of
Agricultural Engineering
University of California



Experimental work in the hot, dry, summer climate of the central valley of mid-California is being performed. It involves a structure, known as a cubicle, which is exposed to the elements in a series of varying conditions, and its consequent thermal behavior observed and recorded. The cubicle is eight feet long, wide, and high, made of stud and plywood construction with mineral wool insulation in its walls and roof-

ceiling. The floor is plywood over a wooden frame and is uninsulated. One full wall is of fixed glass. Thermocouples are placed on the surfaces of each of its faces inside and out and are suspended in the outside air and in the inside or room air. Temperature measurements at each of these points were taken and recorded hourly for several days making it possible to verify the experience of a given day and to average it with that of similar days.

In these tests the glass wall was faced sequentially in eight directions, four cardinal and four intercardinal, in an exposed location with no shade. The thermal behavior of the room air during the daytime period was plotted. Subsequently, attempts were made to protect the glass first by placing a series of materials outside it and then by placing a number inside. In each case the results were recorded and compared with the results obtained for unprotected conditions. Seasonal variations for south exposures were accounted for not only by summer differences (late June and late July) but also by the inclusion of some December work for contrast.

This paper is a progress report giving initial results; additional work has been completed and is being described, and further work is contemplated by the authors.

Load calculations using pretabulated admittance functions

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Dealing with aspects of admittance functions, the first paper of this joint presentation describes a simple and accurate load calculation procedure requiring no special computing machines. The method utilizes information previously obtained with computing machines and is sufficiently flexible to allow the use of local time, variable weather data. The results of the calculations are compared with calculations based on equivalent temperature differentials given in the 1957 ASHAE Guide.

Affecting the solution of a thermal network representing the thermal environment of an enclosure, four variables were studied in the second paper to determine the influence of the formulation of a simplified method of load calculation utilizing pretabulated transfer functions. The factors investigated were the influence of lumping, the importance of sol-air temperature harmonics, the importance of accurately determining the convective conductances, and the effects of absorbed radiation on inside surfaces. These were studied through their effects on the admittance functions of single, composite wall sections.

Large scale wall heat-flow measuring apparatus

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Member ASHRAE

Apparatus was constructed consisting of two 8 x 8 x 4-ft boxes, open on one side, between which the test wall is placed. One box is maintained at the desired constant cold side temperature from –35 to +50 F for steady-state tests or varied according to some predetermined cycle for periodic heat flow tests. The other box is electrically heated to maintain a constant warm side temperature of from 65 to 75 F. The heat transmission coefficient for the wall specimen is calculated from the measured electrical input and temperatures.

Both boxes are lined with an aluminum panel with copper coils attached, through which controlled temperature liquid is circulated. The warm box is heated by the electrical input to the circulating pump motor and heaters contained inside the box. A second set of aluminum panels is installed outside of the warm box liner and separated from it by thermal insulation. This panel is maintained at the same temperature as the liner to guard against any heat transfer through the walls of the box. The cold box is cooled by circulating liquid through the panel liner and finned tubing inside the box. This liquid is cooled in a heat exchanger by low temperature liquid from a central chiller.

In summary, inside box surfaces are controlled at or near air temperature. Natural convection rather than forced convection on the warm side of the test

wall is expected to produce surface conductances close to those occurring in practice.

The heat loss is metered into the whole of the 8 x 8 ft test wall rather than through only a small portion of it. Thus, measurements will be representative of typical heights and whole width modules of the test wall eliminating errors because of the variations in heat flow due to air space convection effects and nonuniformities in the test wall.

The apparatus is expected to provide a tool for assessing the transient response of wall sections and the effects of moisture movement on transient and periodic heat flows.

Heat loss measurements can be made on windows, other building features, and wall sections. Also, with slight modifications, the apparatus can be adapted to general calorimetry work, and for air and water vapor flow measurements. Sources of error in over-all thermal conductance measurements are discussed and the accuracy is conservatively estimated at ±5%.

FIFTH TECHNICAL SESSION—TUESDAY, JUNE 23, 1:30 P.M.

Condensation between the panes of double windows

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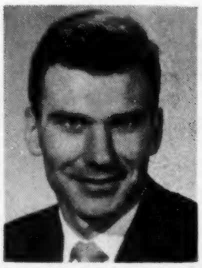
E. S. NOWAK

National Research Council of Canada
Division of Building Research



Condensation of water vapor between the panes, on the inside surface of the outer pane, is a common occurrence during the winter with most types of double windows, except the factory-sealed variety. Such condensation occurs whenever the temperature of the outer pane is below the dew-point temperature of the air-vapor mixture between the panes. This happens, with outside temperatures lower than inside, if the gain in water vapor to the space is greater than the loss. Water vapor moves through the materials and cracks in the window assembly by diffusion because of differences in partial pressure of the water vapor. It is transferred as a component of the air flowing through cracks under total pressure differences.

The vapor pressure difference across the inner pane is likely to be several times greater than the vapor pressure difference across the outer pane, the ratio of these pressure differences increasing with decreasing outside temperature. Thus, with vapor transfer by diffusion alone, the effective resistance of the inner glazing to vapor flow by diffusion must be many times that of the outer glazing in order to maintain outflow equal to inflow.

The air space can interchange air with the outside when openings around the outside pane are located at the top and bottom of the window, even with a total pressure difference across the window and an overall air flow from inside to out, provided that the air-flow resistance of the inside pane is sufficiently higher than that of the outside pane. A simple mass balance indicates that, with vapor transfer by air flow alone, the amount of air flow from outside into the air space must be several times that from the inside into the air space to prevent condensation on the outside pane, the ratio increasing with decreasing outside temperature.

Vapor transfer by diffusion is compared with that by air flow for a specific window from measured diffusion and air leakage characteristics, and fair agreement is obtained with observations of window condensation in a cold room installation. Buoyancy forces, providing air interchange between the window space and outside, most effectively control condensation. The venting required to prevent condensation for given design conditions can be predicted from simple pressure relationships, providing inner pane air flow data are available. Excessive air interchange between the air space and outside can increase significantly the over-all heat transmission coefficient, making venting impractical for windows having high leakage rates.

Evaporative cooling for common storage of fruits and vegetables

Fruits and vegetables harvested in the fall are stored in large quantities to prevent market gluts and to provide a regulated supply for the winter and spring months. Storage facilities are refrigerated or air cooled. The latter type of storage is used almost exclusively for crops such as potatoes. Growers avail

themselves of air cooled storages for the other fruits and vegetables if the time in storage is limited.

Air cooled or common storages in the early fall may suffer from high temperatures that encourage disease and respiration losses. There is also no control over humidity. Through the use of the evaporative cooling principle, it is possible to improve common storage conditions. Temperatures are lowered, thereby reducing produce losses. Humidity is raised, thus minimizing shrinkage and shriveling. The fresh washed outside air, introduced by the cooling system, also expels ripening gases and odors often associated with common storages.

The paper briefly reviews the principles of produce physiology and environmental conditions needed for the storage of fruits and vegetables. It then outlines the mechanics of evaporative cooling. Typical examples are presented with design data given for laying out evaporative cooling systems for common storages of apples and potatoes. Also included is information on citrus cooling and the coloring of oranges, grapefruit and lemons.

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Controlled atmosphere apple storage process and its requirements on refrigeration structures and systems

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Reporting special considerations in the design of refrigeration systems and structures to insure compatibility with C-A apple storage process requirements, it is found that good operating procedures for the C-A rooms involve the use of nitrogen, a breather bag, and a CO_2 absorption tower in bringing the process through the initial transient state conditions.

The satisfactory performance of the C-A storage can be predicted by means of gross gas permeance tests (by means of a pressure relaxation curve) after completion of all construction and installation of all equipment. The relationships between size and heat of transmission, sensible heat and heat of respiration are presented, based on: (a) 10 day loading rate to capacity, (b) 24 hr cooling to 32 F, (c) ceiling U value of .05, (d) wall U value of .075, and (e) floor U value of .15.

High air velocities, essential during initial cooling loads can later result in unnecessary heat loads on the storage and in some cases excessive infiltration of outside air into the storage due to the Bernoulli effect of high velocities across any pin hole leaks in the storage gas seal.

SIXTH TECHNICAL SESSION—TUESDAY, JUNE 23, 1:30 P.M.

Noise in refrigeration and air conditioning systems

C. M. ASHLEY
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Carrier Corp.
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The Joint ASRE-ASHAE Committee on Equipment Sound Testing was set up in 1958 to provide an essential step in the progress toward better industry wide equipment sound control. As background, the committee had a general ASA sound measurement standard, and much specific research and testing of equipment sound. Sound power level was chosen as the unit of rating to avoid the effect of the specific environment, and octave-band measurements were used to indicate the character of the noise.

Direct determination of the sound power level of the equipment being measured, by comparison of sound pressure levels with a calibrated standard sound

source, is an important tool to improve the simplicity and accuracy of measurement. Readings are taken in a reverberant room in which sound is averaged by multiple reflections, thus avoiding the complications of a multi-station traverse. The use of such a room is suitable, since air conditioning and refrigerating equipment noise is generally broad band in character and directivity effects are not usually of great importance.

One standard is expected to cover self-contained equipment, in which the sound radiates directly into the air, possibly with sub-standards for equipment such as room air conditioners. Another standard is planned for the measurement of the sound output of ducted fans, and a third standard for measurement of sound generation, attenuation of outlets, and other terminal devices attached to ducts. The program can be expanded further in the future.

Experimental study of grille noise characteristics

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Tennessee Valley Authority
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Seeking to determine how grille noise characteristics are affected by fin design, static pressure loss, and grille face velocity, this study uses three test grilles, of the return-air type, having different fin shapes and spacing for the experiments. The grille noise characteristics of each grille are compared on the basis of octave-band spectra, subjective loudness spectra, and total loudness. The test apparatus consists of a controlled air supply system, sound trap, and a small anechoic-type chamber, in which the grille noise intensities are measured.

Total loudness curves of the grille noise, which are evaluated from the loudness spectra, are also shown. A qualitative analysis of the graphical results shows that fin design, static pressure loss, and grille face velocity are interrelated factors which affect the frequency and loudness characteristics of grille noise in varying degrees. The three test grilles are rated on the basis of total loudness evaluation.

Octave-band analyses are made at various air flow rates to obtain the total and background grille noise data for each test grille. The octave-band sound-pressure level data of total noise are corrected to obtain octave-band sound-pressure levels of grille noise per square foot of core area. The octave-band spectra for each grille are graphically presented as a function of grille face velocity. The octave-band spectra values are converted to sones to produce graphs representing the subjective loudness spectra of the grille noise.

Effect of fuel composition on oil burner noise

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Member ASHRAE

W. A. BEACH
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H. F. SCHROEDER
Process Research Div
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and Engineering Div

Representing a segment of a broader program on domestic oil burner noise suppression, the effect of fuel composition on oil burner flame noise was studied.

A typical warm air furnace-burner combination was used for the test work. It was installed in a laboratory which had an extremely low background noise level, the equivalent of a radio broadcasting studio. The sound measuring instruments used gave results reproducible within one decibel. All combustion variables were held constant throughout the test except for fuel composition.

Nine different fuels were tested for their effect on the largest oil burner noise contributor, the flame. These test fuels included typical products now mar-

keted, fuels composed entirely of cracked products and several pure compounds. Despite this wide variation in fuel composition, there was no significant difference in the overall noise level, or in the noise level in any of the individual octave bands with fuels boiling in the range of kerosenes and No. 2 home heating oil. A fuel composed of 100% catalytically cracked oil, a fuel containing 100% thermally cracked oil, and a straight run virgin kerosene, all gave the same sound spectrum as regular No. 2 heating oil. Of all the tests made, only the low boiling paraffins, normal heptane and isooctane, showed noise levels that varied significantly from the other fuels tested. These pure compounds are found in gasolines and are not used as fuels for regulation type oil burners.

From these studies we conclude that fuel composition is not a significant factor in the noise produced in domestic oil heating units. Changing fuel composition, therefore, does not appear to be a method by which oil burner noise can be reduced.

SEVENTH TECHNICAL SESSION—WEDNESDAY, JUNE 24, 9:00 A.M.

A comparison of fluoroalkane absorption refrigerants

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Member ASHRAE

Utilizing fuel directly for cooling, absorption refrigeration systems are being used by industry, and improved upon by research. Many absorption systems are deficient in various practical ways. For example, water cooling to achieve low pressures is required in the water-lithium bromide system and, also, lithium bromide is corrosive. Ammonia-water systems can be used with an air cooled condenser, but ammonia is flammable and toxic. Therefore, indirect cooling, through chilled water or brine is used, although far from ideal. Water is too volatile as an absorbent, and the system pressure too great. A fluoroalkane refrigerant, however, operates with air cooled condensers, and being non-flammable and practically non-toxic, can be used to cool peopled areas.

Comparing six fluoroalkane refrigerants on a ratio basis, one with another and with methylene chloride, the author assesses their relative merits, independent of the particular machine used.

Chemical stability is of prime importance for any system, and was determined by sealing the refrigerant and absorbent in glass pressure tubes containing metals of construction, and storing them at high temperatures for weeks or months. The extent of decomposition was gaged by appearance, penetration of the metal strips and chloride ion formation. Most stable was Refrigerant-22, which shows good chemical and thermodynamical properties, although deficient in having a relatively high operating pressure. Halogenated alkane refrigerants could not be compared directly with ammonia or water as refrigerants because of operational and apparatus differences. Still, without a highly accurate comparison, the two inorganic refrigerants show high latent heats and good solubility characteristics, thus, relatively high performance coefficients. Not, however, as high as Refrigerant-22. This, the preferred system, has enough advantages to merit careful consideration for all absorption refrigeration requirements.

Estimating water content of certain refrigerating compressors

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Refrigerator-Freezer Engineering
Westinghouse Electric Corp.

H. M. ELSEY

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Member ASHRAE

J. B. KELLEY

Process Engineer
Refrigerator-Freezer Engineering
Westinghouse Electric Corp.

Member ASHRAE

Investigating several methods of estimating the water content of dried but uncharged hermetic refrigerator compressors, the study was to select the method which would give an approximate answer as soon as possible after a compressor was removed from the dehydration oven.

The most satisfactory of the procedures studied

is as follows: Weld the head on the compressor shell immediately after it leaves the oven and then place it in a small oven which is already at 300 F. Evacuate for 45 min through a moisture collecting device.

Two moisture collecting devices have been used. The presently preferred one is a cold trap which is designed so that, after it has been warmed to room temperature, it can be centrifuged. The volume of the water which has been collected can then be read from the scale on the capillary tube which forms the bottom enclosure of the moisture-collecting apparatus.

An alternate moisture collector is a lightweight metal drier which is charged with Molecular Sieve 4A. Other gases which are also absorbed and held tightly by this desiccant are either absent or present in such small amounts that the moisture determinations are valid for quality control purposes.

The role of water in Refrigerant-12 systems and the problem of dehydrating systems containing stators which are insulated with cellulose are also discussed.

Viscosity of refrigerants in the vapor phase

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Purdue University

C. Z. KAMIEN

Instructor in the School of Mechanical Engineering
Purdue University

Using a rolling-ball viscometer, measurement has been made of the viscosities of some fluorinated hydrocarbon vapors. The viscometer consists of a ball rolling down an inclined tube filled with the fluid under investigation. The variables which affect the ball velocity in such a tube are also investigated, and certain equations which are employed in the calculations are introduced.

It was found that the effect of temperature changes on the calibration constant is appreciable when high temperatures are encountered or when the materials used have high coefficients of expansion. Therefore the values of the calibration coefficient had to be determined whenever temperature changes, physical changes of apparatus, or reassembling occurred.

The literature on the determination of the viscosity of various gases other than the fluorinated hydrocarbons is reviewed at some length. Also, a detailed description of the apparatus and the calibrations is made. The results of the investigation are shown graphically in the figures, where it is possible to see the relative viscosities of the refrigerants under question, and the effects of the important variables on the refrigerants. Further work is being done to assess the viscosities of the same refrigerants under a different set of variables. There is a need for further work along these lines in other of the refrigerant families.

Repeated scrape abrasion testing of enameled wires in frozen and liquid refrigerants

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Westinghouse Research Laboratories

G. J. BICH

Insulation Department
Westinghouse Research Laboratories

G. W. HEWITT

Insulation Department
Westinghouse Research Laboratories

A new tester has been designed for measuring the effects of gaseous and liquid refrigerants on the scrape abrasion resistance of enameled magnet wires intended for hermetic use. With this tester, the specimens are scraped while immersed in the refrigerant gas or liquid, simulating actual conditions of operation in hermetic systems.

Advantages of this newly designed tester are that 48 tests can be made on 16 wire specimens in a single charging of the tester, and that the load can be changed from outside the tank under full refrigerant pressure. Each of the 16 wire specimens is scraped on three areas on the same side of the wire. This provides for testing the same wire specimen under

three conditions such as air, refrigerant gas and refrigerant liquid. A description and photographs of the tester are given in this paper.

Various chemically different types of wire enamels have been tested in this unit in Refrigerant-12, Refrigerant-22 and air. These include the following: phenolic modified polyvinylacetal (Formvar), polyacrylonitrile, epoxy modified polyester, polyamide, amine catalyzed epoxy, Formvar modified epoxy, polyesteramide, and polyurethane modified polyesteramide.

Of the enameled wires tested, only the polyamide and the polyurethane modified polyesteramide were not affected adversely by Refrigerant-22 liquid after more than 20 hr immersion. The enamels most severely affected were Formvar and the Formvar modified epoxy.

With the exception of Formvar, results of tests in Refrigerant-12 gas, in contrast with those in Refrigerant-22 gas, show improvement in the repeated scrape abrasion resistance, for the enamels tested to date.

EIGHTH TECHNICAL SESSION—WEDNESDAY, JUNE 24, 9:00 A.M.

Determination of the effectiveness of window shading materials on the reduction of solar radiation heat gain

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Member ASHRAE

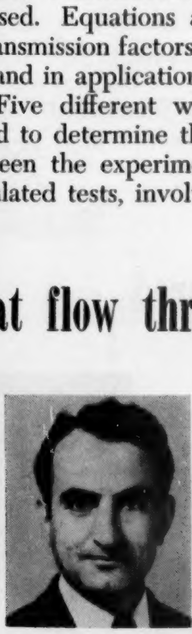
J. L. THRELKELD
Professor
Department of Engineering
University of Minnesota
Member ASHRAE

Reporting the procedures by which window shading material transmission factors may be determined, the authors present the solar calorimeter tests, and calculations, involving independently measured reflectivities and transmissivities of the materials which may be used. Equations are derived for the calculation of transmission factors both in application to calorimeters and in application to windows in rooms.

Five different window shading materials were tested to determine the transmission factors. Checks between the experimental calorimeter tests and the calculated tests, involving measured reflectivities and transmissivities, showed excellent correlation. In most cases the calculated determinations can be used with reasonable accuracy, particularly in those cases where the physical constants of the materials can be reliably evaluated.

The sensitivity, however, of the calculated transmission factors to determinations of the reflectivities and transmissivities is such that the experimental variations may materially affect the overall transmission values. In those cases where the physical constants may be open to question, calorimeter tests are desirable.

Heat flow through glass with roller shades



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ASHRAE Research Laboratory

L. F. SCHUTRUM
Research Supervisor
ASHRAE Research Laboratory
Member ASHRAE

In this study, summer heat gains and winter heat losses through a glass roller shade combination were investigated for shades of different materials and colors, as well as for different type glasses and various glass shades. Included are data related to heat gains for typical combination in East, South and West orientation, and for various hours of the day.

Drawing certain conclusions from the tests made, and the data accumulated, the authors state that both summer heat gains and winter heat losses are calculable. Also, it was found possible to determine the shade temperatures independently of the outside temperature. With an opaque roller-shade, solar heat gains are reduced by lowering the solar absorptance of the surface of the shade on the glass side.

The effect of a low emissivity surface for long wave radiation on the room side of the shade is to reduce the heat gain due to solar radiation. The effect, however, of a low emissivity surface on the glass side of the shade is to increase the solar heat gains with a regular plate glass, and to decrease it with a heat absorbing glass. Furthermore, with an opaque shade, having a white surface on the glass side, the solar heat gains are less with regular plate glass than with heat absorbing glass. With a dark green shade, however, heat gains are less with heat absorbing glass than with regular plate glass.

Evaluation of three room air distribution systems for summer cooling

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A. KOESTEL

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G. L. TUVE

Professor of Mechanical Engineering
Case Institute of Technology

Member ASHRAE

Three systems of room air distribution for summer were evaluated on the basis of specific criteria for comfort. A theory of room air motion and a philosophy for evaluating comfort conditions are discussed. Extensive tests were made in a room equivalent to a well insulated living room with exposed walls, operating with surface temperatures simulating outdoor conditions up to 100 F. The high sidewall air supply system would meet the rigid comfort requirements at moderate loads and low supply volumes only. With baseboard radial diffusers, drafts in some parts of the room could be eliminated. The system using round ceiling diffusers was more satisfactory than either of the others. A comparison is made with earlier tests of all three systems for winter heating under various conditions of operation.

It was found that the air flow pattern was independent of flow rate, and was excellent for air distribution and that it made full use of the unoccupied zone for air entrainment and load absorption. Limitations of room size made it necessary to keep the flow rate down to ten air changes per hour or less in order to prevent drafts at the six foot level, midway between the two diffusers. But the system satisfied all comfort criteria for summer cooling when operated at lower flow rates.

Surface odor adsorption and retention properties measured

W. F. HOPPER

Research Engineer
ASHRAE Research Laboratory

Presenting the results of a study made to determine a suitable method of measuring the odor adsorption and retention characteristics of various surfaces, the author proposes a method involving an application of the syringe technique. By this technique odorous and odor free air is mixed to obtain threshold level as judged by an organoleptic panel.

In studying the suitability of the test method some data on the odor adsorption and retention properties of wool, cotton, nylon and rayon fabrics were obtained at several ambient temperatures and relative humidities.

Data based on tests of samples cut from a single bolt of each of the three fabrics show that the odor adsorption during a 24 hr period was higher for nylon and cotton than for wool, and that the maximum adsorption rates occurred at about 75 F and 50% relative humidity. Odor decay curves for these three fabrics show about the same rate of odor release by nylon and cotton, and a less rapid release by the wool. The odor absorption by the rayon samples was so low that it could not be detected by the method employed.

CONFERENCES

to review problems and progress in Domestic Refrigerator Engineering Industrial Ventilation Cryogenics

DOMESTIC REFRIGERATOR ENGINEERING CONFERENCE, MONDAY, JUNE 22, 9:30 A.M.

F. A. NOLL
Presiding Chairman

Design and basic functions required of present day domestic refrigerators will be discussed at the Domestic Refrigerator Conference. Speakers will present papers on the methods of controlling temperature, effects of humidity and temperature conditions, and cooling by forced air movement in the freezing section as well as in the fresh food section. The Conference will help keep us from overlooking the basic design requirements to make certain that the question, "Should the Ice Box Return?", will never have to be asked.

Keynoter—

COLONEL C. S. LAWRENCE Executive Secretary, Institute of Food Technologists

Control of temperature

E. VON ARB
Revco Inc.
Member ASHRAE

Presenting an unusual method of controlling food storage temperature, the author shows how to obtain the minimum temperature variation from a preferred setting, and also a convenient choice of temperature, with assurance of maintaining it regardless of location. The problems encountered in the development of this system, and their solutions, are also discussed.

Humidity and temperature conditions

Storage time of fresh foods in a domestic refrigerator is directly related to cabinet temperatures and humidities and in this conference the effects of refrigeration temperatures and humidities on the dehydration rates of foods and the growth rate of food micro-organisms are discussed. Also, the storage time of selected foods in a theoretically determined "ideal" atmosphere is compared to the storage time in contemporary domestic refrigerators. The engineering and design of the "ideal" domestic refrigerator for food preservation is discussed.

E. W. ZEARFOSS
Development Engineer
Philco Corporation
Member ASHRAE

F. P. SPEICHER
Biologist
Philco Corporation

Air circulating in freezer compartment—benefits and problems

C. H. WURTZ

Manager of Refrigerated Appl. Engineering Frigidaire Div General Motors

The removal of heat from the frozen storage compartment of the refrigerator by circulation of low temperature air, presents both new problems and new features for the industry. For this reason, the design factors of air flow relative to evaporator surface and heat load will be discussed. The benefits of cooling and sublimation within the frozen storage compartment will be shown by slides and charts and compared to the conventional type of evaporator used in similar service.

INDUSTRIAL VENTILATING CONFERENCE, TUESDAY, JUNE 23, 9:00 A.M.

G. B. PRIESTER

Presiding Chairman

Four papers are to be presented at this Conference dealing with methods and problems involved in providing more comfort for industrial workers. As it has been stated many times before, improved worker environment can increase worker efficiency by making working conditions more desirable.

The first speaker will discuss the importance of industrial supply air systems, pointing out that consideration must be given to the structure of the building and the manufacturing procedures and processes involved. The physical aspects of ventilation and air conditioning in industry will be mentioned. The design requirements of the equipment and air distribution system, including supply air outlet types and their uses, will be covered.

The second speaker will talk more specifically on dust and fume control systems, citing the need for these and then giving details on the design of systems including pictures of a specific installation.

The third speaker will discuss ventilation in the steel industry, using slides to illustrate the progress made during the past twenty years. Detailed descriptions of a small steel company's heating and ventilating systems will be given, including a detailed account of an unusual locker room ventilation system. He will state that even as employees must have a heated plant during the winter they must have a cooled environment during the summer, and he will report upon his company's approach to this problem.

The fourth speaker will discuss control of radiant heat in factories to provide more comfortable working conditions.

Design and performance of industrial supply air systems

J. H. CLARKE

Visking Corporation
Div of Union Carbide

Industrial supply air systems have become increasingly important because of modern plant design, substantially increased internal heat loads, greatly increased makeup air requirements, and greater emphasis on efficiency. Industrial air conditioning and relief ventilation are good investments resulting in higher grade employees, safer working conditions, less absenteeism, less labor turnover, and proven increases in production and product quality of substantial amounts.

To permit efficient air conditioning and ventilation, the system and equipment requirements must be considered in the design of the building, process and process equipment. High human efficiency is obtained either by complete air conditioning, or by relief ventilation systems, which minimize body stress by providing adequate convective or sweat evaporative cooling within prescribed limits.

The system components must be designed for reliability, effectiveness and ease of maintenance. Of particular importance for makeup systems is freeze protection. This is provided by correct design and selection of the

system casings, dampers, heaters, fans, controls and piping.

Package air handling units have been used in large numbers in industrial applications. They have often been oversold and have failed to measure up to the necessary quality and performance standards. Prefabricated systems, with quality components selected for efficiency and the job requirements, should be carefully considered for the economies possible.

Air distribution systems vary widely from the minimum or no duct work requirements of a simple makeup system which lets air out into a space, to the elaborate systems required for relief ventilation or makeup at high air change rates. Distribution in fully air conditioned areas can be conventional. For efficient heating and relief ventilation, the air should be supplied to the work zones. Special outlets designed for the specific ventilation requirements should be provided. Directional or volume adjustment of the outlets is essential for relief ventilation to permit the worker to regulate the desired ambient air motion, maintain the required makeup air volumes and permit economical tempering of the supply air. In hooded and process areas, means must frequently be provided to introduce the supply air without a disturbing motion at the hoods or product.

Dust and fume control systems

P. J. MARSCHALL
Chief Mechanical Engineer
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General ventilating in the steel industry

F. E. TUCKER
Industrial Hygienist
Weirton Steel Company

Evolution of industrial ventilation at the Weirton Steel Company, W. Va., has lead to the installation of an area heating and ventilation system which supplies conditioned makeup air to plant buildings at a rate of two cu ft per min per sq ft of floor space. This air is distributed through a series of discharge outlets designed to deliver the air at an elevation of 10 to 12 ft above the floor.

This discussion includes the ventilation required for a new centralized locker room located in an area of high heat load inside a plant building at an elevation of 40 ft above the floor. It is designed to accommodate 2,000 employees; all air is brought in through a large plenum located beneath the floor and discharged through grilles located beneath each locker base. There are a total of over 100 discharge outlets from a single plenum located beneath the floor.

A prediction of what the steel industry can expect in the future in the way of industrial ventilation demands is made. This portion of the paper also discusses present air-conditioning practices in crane cabs and control pulpits.

Radiant heat control in industrial plants

The methods of controlling radiated heat to provide comfortable working conditions in industry are totally different from the means used to control air temperature and humidity. Sources of unwanted radiated heat are not at all uncommon; but their influence is invariably underestimated, or completely neglected, even though they may make up most of a person's external heat load.

W. G. HAZARD
Industrial Relations Div
Owens-Illinois

How to measure radiated heat, how to evaluate it comfort-wise, and how to control it, will be discussed, and illustrated with simple demonstration apparatus.

In addition to an unusually extensive and broad Technical Program which includes those 29 technical discussions, 11 conference papers and 6 forums indicated upon this and accompanying pages, the annual meeting of ASHRAE in Lake Placid, N. Y., will feature such other special events as the Welcome Luncheon (with installation of new officers), trips to Whiteface Mountain and other points of local interest, a chair-lift ride at Ski Chalet, an outdoor steak fry, cocktail parties and the customary Banquet with the presentation of awards and honors.

V. J. JOHNSON
Presiding Chairman

The rapid increase in use of liquefied gases for industrial, defense, and space probe purposes has necessitated a scientific and engineering approach to the production and handling of a variety of cryogenic fluids. The primary fluids of this class are air, oxygen, nitrogen, hydrogen and helium. Other cryogenic fluids receiving attention are neon, argon, fluorine, methane and carbon monoxide. Liquid oxygen perhaps has been used industrially in large quantities longer than any of the others, dating back to the early forties for increasing steel production. It is now used in much greater quantities as a propellant component for rockets. Liquid hydrogen is also being used in greater and greater quantities for the development of propulsion systems. Liquid helium (as well as most of the cryogenic fluids) can be transported more cheaply in liquid form and so are being liquefied for transportation even though they are to be used as a gas.

The many phases of this fast growing new industry of Cryogenics has encompassed a wide field of technology and now includes several thousand engineers, scientists and technologists, many in the refrigeration and air conditioning industries. Four annual or bi-annual cryogenic engineering conferences have been held with attendance record of over 500. The fifth one, scheduled for early September this year at the University of California, is expected to attract 600 to 700 people.

The more popular aspects of handling cryogenic fluids have been selected for this ASHRAE Conference for particular interest to refrigeration and air conditioning engineers. Pumping, transferring, storing, flow measurement, liquid level detection, pressure drop, and temperature measurement will be discussed along with the design features of pertinent equipment. The abstracts of prepared papers refer to specific subject coverage to be presented. Additional time will be provided for general discussion of any phases of closed circuit transportation of liquefied gases that are of particular interest.

Transfer of cryogenic fluids

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Discussing the present status of equipment and techniques for the transfer of cryogenic fluids through piping systems, this paper presents a summary of some calculation procedures.

The topics to be discussed include properties of materials, insulation (uninsulated and conventionally insulated systems as well as a few remarks on vacuum insulated systems), single-phase and two-phase flow considerations, the cool-down problem, pumps, heat transfer, instrumentation, fittings and valves, and safety.

It is stressed that cryogenic engineering is essentially a synthesis of the older branches of engineering, and therefore cryogenic problems are not basically different from the more familiar problems. However, it is also pointed out some new techniques have been developed, the emphasis for some problems is different, and there is a serious lack of information in several important areas.

Cryogenic pump design

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Pesco Products
Div of Borg Warner Corp.

A brief summary of the advantages and disadvantages of using cryogenic fuels and oxidizers will be followed by a discussion of the mechanical and hydraulic design considerations for cryogenic pumps. Special attention will be given to material choice, bearing and seal problems and cryogenic pump cavitation performance. The major part of the discussion will be devoted to liquid hydrogen, and for this reason the properties of this cryogenic liquid will be discussed with relation to handling and safety procedures.

The specific experience gained in the design development of a submerged, electric motor driven, centrifugal, liquid hydrogen transfer pump will be used to highlight the areas where particular design care must be exercised. The facilities and instrumentation needed for the experimental evaluation of the pump will be discussed and the pump and submerged electric motor performance will be presented. Particular attention will be given to the exceptional cavitation performance obtained with the pump, and the weight saving which can be realized by operating electric motors at cryogenic temperatures.

Cryogenic insulation materials and techniques

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Cryogenic Engineering Laboratory
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Methods of insulating vessels for the transport and storage of liquefied gases will be discussed. A new technique of separating multiple radiation shields in vacuum using glass paper has been investigated. The effective thermal conductivity between 70 F and –423 F is less than 3 x 10^{-5} Btu hr^{-1} ft^{-1} F^{-1}, or about 10 times better than the best evacuated powder insulations. Use of this type of insulation is especially advantageous where pay load is highly important, for example, the transport of liquid helium. In addition, the conductivity of a wide variety of evacuated powders will be presented. Emphasis will be placed on silica powders mixed with metal powders, either aluminum or copper. The addition of 20 wt. percent of aluminum to silica aerogel can reduce the effective conductivity to ¼ its original value.

The use of high vacuum insulation for transfer lines and Dewar vessels will be discussed. Representative emissivity data will be given. Carbon dioxide emitted into the vacuum space at 1.5 atmospheres pressure will condense when the transfer line or vessel is cooled. This technique greatly lengthens the storage time of transfer lines, because the insulation space contains a positive pressure of carbon dioxide while it is at ambient temperature.

The heat conduction through the supports of a vessel can usually be reduced to the same order of magnitude or less than other thermal losses. The thermal conductivity and mechanical properties of some selected support materials have been tabulated.

Instrumentation of cryogenic research installations

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At the Lewis Research Center hydrogen, oxygen, fluorine, and nitrogen are the primary fluids used in research on rocket systems. The measurements taken vary only slightly, whether the concern be with tankage, pumps, gas generators, turbine or rocket motor. In each case the concern is with mass flow, pressures, temperatures, and probably one or more parameters common only to the component, such as thrust, speed, or torque. This paper will discuss how the requirements of the instrumentation vary with component, cryogenic fluid, and test requirements.

Conventional transducers can be used to get most of the data, however, for large scale testing, the low temperatures and fluids cause some new problems. These systems will be discussed. Factors which must be considered in the choice of instrument transducers and components which constitute the system are temperature, pressure, heat transfer, length of test, compatibility of materials, safety and environmental conditions of the test. From these factors it can be seen that instrumentation must be considered early in the design of the experiment. A review of applicable transducers will be given.

FORUMS

Forums proved to be one of the more popular features of predecessor ASRE national meetings. They offer highly informal discussions for the exchange of ideas. There are no prepared papers or talks, no notes or minutes are taken, it is all "off the record." Our guests of the press respect the spirit of the forum. Here is an excellent opportunity to exchange and discuss ideas and opinions.

Six diversified subjects will be covered at the ASHRAE Lake Placid annual meeting. These were selected on the basis of popular interest expressed from various sources. The forums to be presented are as follows:

Status of new utilization voltage proposals for air-conditioning equipment
—Moderator, A. S. Anderson, Electrical Engineer, Ebasco Services

Icemaker design considerations
—Moderator, E. MacLeod, Engineer, Carrier Corporation

Research, the engineer, and the appliance industry
—Moderator, E. A. Baillif, Chief Engineer, Refrigerators, Whirlpool Corporation

Immersion chilling of poultry
—Moderator, E. N. Kerrigan, Manager, York Corporation

Sound and vibration problems in air delivery equipment
—Moderator, J. B. Graham, Director of Research, Buffalo Forge Company

Freeze protection of heating coils
—Moderator, Earl Wilson, Supervisory Mechanical Engineer, Abbott Laboratories

Nomograph design

and selection of counterflow towers

GEORGE W. BROOKE

During the past 15 years the cooling tower has assumed a position of considerable importance in refrigeration and cooling systems. Increasing water demands of a growing population and industry have destroyed the myth that our water supply is inexhaustible. Since water supply has drawn great interest from those who have need of cooling systems, there has been a corresponding interest in methods of re-using water that is now discharged as waste. Cooling towers use approx. 3 % make-up water while 97 % of the water is used again. The make-up water is necessary to replace the portion of water which evaporates during the tower cooling cycle, evaporation being the vehicle by which the remaining water is cooled. All points considered, the cooling tower is an efficient instrument for conserving water while performing an important industrial function.

The design of cooling towers to meet particular industrial applications has presented something of an enigma for a number of years. Until recently the most popular design method was that based on differences in air and water specific enthalpies. During the past few years there has been growing interest in the possibility of tower design based on mean cooling potential.

F. Rasori* pointed out the relative merits of the two tower design methods. Subsequently, Mr. Eduljee and Mr. Raman** modified the cooling potential method to the point where tedious mechanical integration was no longer necessary; the modified method is still rather unwieldy, however.

With this latter fact in mind the author has tried to construct a rapid, yet accurate, method of cooling tower design. The method which presented itself as being most logical was that which might incorporate the modified cooling potential method in the form of charts of mathematical design, i.e., nomographs. A tower design method based on cooling potential utilizes a total heat transfer design coefficient, K_a, as defined in the table of terms. Each tower manufacturer has units which are capable of functioning over a wide range of air and water temperatures. Thus a particular tower can operate to meet a wide range of required K_a values.

Fig. 1

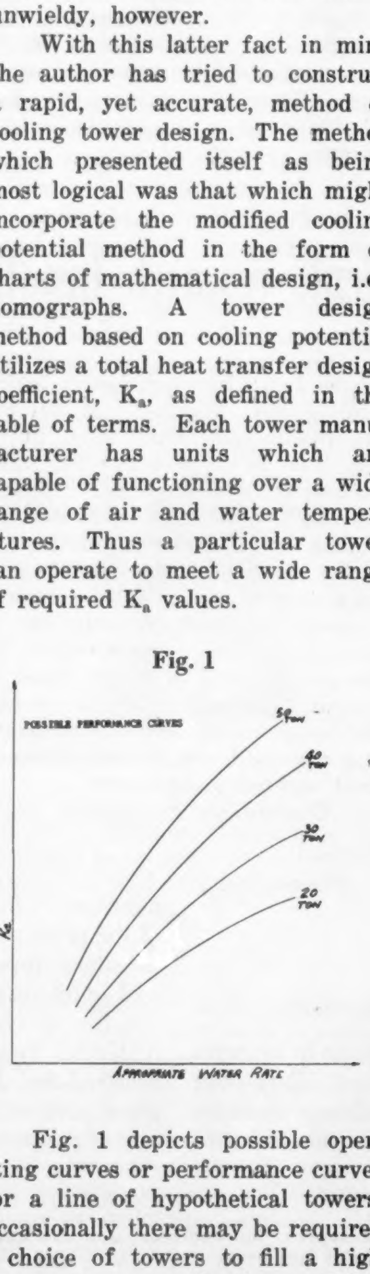

Fig. 1 depicts possible operating curves or performance curves for a line of hypothetical towers. Occasionally there may be required a choice of towers to fill a high temperature application. High temperatures are frequently attendant to motors, compressors, heat treating equipment, plating baths and process equipment, to mention a few familiar examples.

In applications of this type the overall heat transfer coefficient, K_a, will be high. A high value of K_a would necessitate the use of a larger tower or pack volume or possibly a greater air flow rate. If the engineer decides to use a particular tower for an application (space limitations may proscribe the use of a tower beyond a certain size) it may be necessary to increase the air rate, assuming the required K_a is quite high.

The problem below has been chosen to point out the inherent advantages of using nomographs in the solution of tower design problems. The use of nomographs can be time saving, particularly where repeated tower design solutions are desired. The choice of limits for the variables used is an important consideration and must be dependent upon the towers of the company for which the design method is being formulated. One company may use small pack volumes and high air flow rates while another might use the opposite arrangement.

Problem: It is desired to choose a tower which will cool 300 gal of water per min from an entering temperature, T_1, of 96 F to a leaving temperature, T_2, of 87 F. (These figures are chosen at random, merely to apply to this hypothetical problem.) We wish to use a tower with a flow rate of, say, 17,500 cu ft of air per min and find there is a tower available which has an air rate of from 15,000 to 20,000 cfm (it is possible to vary the air flow rate by adjusting the fan sheaves). The

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*F. Rasori, "Cooling Potential versus Specific Enthalpy", *Refrigerating Engineering*, p. 1207, November, 1953.

**H. E. Eduljee and B. V. Raman, "Simplifying the Cooling Potential Concept As Expressed by F. Rasori", *Refrigerating Engineering*, p. 76, June, 1954.

George W. Brooke is Systems and Procedure Analyst, Detroit Controls Div, American-Standard. Original study made while an Application Engineer with Acme Industries. This paper was presented at the 45th Semiannual Meeting of ASRE in New Orleans, La., December 1-3, 1958.

entering air wet bulb temperature, t_a, is 73 F.

Step 1. Using Fig. 2 draw a straight line from 300 gpm through 17,500 cfm to pivot line A. From the intersection on line A draw another line through the 9° water temperature drop (96° — 87° = 9°) to the second pivot line B. From this second intersection draw a third line through the air-in wet bulb temperature (t_a = 73 F) to the air-out temperature scale. Read the temperature as t_1; the air leaving the tower has a temperature of 87 F in this example.

Step 2. Table 1, Appendix, lists the enthalpy values of air at various temperatures. Note the enthalpy values corresponding to the air-in and air-out temperatures, t_a and t_1 respectively. In order to avoid the mechanical integration of the type demonstrated in the article by Mr. Rasori it is necessary to use what is known as Simpson's rule for areas.* To the air-in enthalpy (refer to Table 1) add half the difference between the two extreme values of air enthalpy. These are the values 36.7 and 51.9 corresponding to t_a and t_1. The resulting enthalpy corresponds to the mean air temperature, t_2. To find the water temperature associated with the mean air temperature take the arithmetic average of the water-in temperature, T_1, and the water-out temperature, T_3. The average of 96 F and 87 F is, of course, 91.5 F.

	t_a	h_a
1	87	51.9
3	73	36.7

$$dh_a = \Delta h_a = 15.2$$

$$h_{a2} = dh_a/2 + h_{a3}$$

$$= 7.6 + 36.7$$

$$= 44.3$$

$$t_2 = 80.1 \text{ F}$$

Step 3. The air and water temperatures have been related and Fig. 3 can be used now to find the "g" factor in each case. Draw a line through T_1 and t_1 intersecting the "g" factor scale; the "g" value is about 2.15 in this case. Repeat this same procedure for the other air and water temperatures.

Step 4. It is now possible to use Simpson's rule to find the "G" value for the tower. To do this it is necessary to multiply the "g" factor which corresponds to T_2 and t_2 by a factor of four:

$$1.8 \times 4 = 7.2$$

To this value add the "g" factors which correspond to the T_1 — t_1 and T_3 — t_3 temperatures:

$$7.2 + 2.15 + 1.50 = 10.85$$

Divide this total by six; the resulting value, 1.81, is the mean cooling potential difference.

Step 5. It is now possible to determine the K_a value required of the tower by the use of Fig. 4. This is found to be about 63,000.

*For a discussion of the use of Simpson's rule see the Appendix.

It is thus necessary for the tower to reach a K_a value of 63,000 in order to meet the required load —cooling 300 gal of water per min from 96 F to 87 F with an air rate of 17,500 cfm. A given tower must be designed to meet the possible K_a requirements to be met in industrial applications and curves must be constructed similar to those of Fig. 1 by the tower manufacturer. In making tower applications the same design procedure as that of the example is followed; then, knowing the performance of different models of towers and their air rates and pack volumes it is possible to predict the performance of new towers (with the same type pack).

Often, when designing towers to supply a given K_a under a particular set of load conditions, it is necessary to make a number of trial solutions by the older method (that pointed out by Eduljee and Raman). Using a nomograph such as the one shown in Fig. 4 will make it possible to visually juggle different values for the tower design variables such as volume of pack or water temperature drop. This will reduce the time required to make tower applications with existing units and to design new units following the same procedure.

APPENDIX

Terms

GPM—Water rate desired or amount of water being cooled, measured in gal per min.

Volume of Pack—The total volume within which heat transfer will normally occur. Though there may be some transfer outside the pack proper this is probably small enough to be ignored.

"G" factor—the reciprocal of the mean cooling potential difference as determined by Simpson's rule for areas. This is a simplified term for the factor

$$\left(\frac{1}{\theta w - \theta a}\right)_m$$

K_a—Total heat transfer coefficient, Btu per hour per cu ft per in. of mean cooling potential difference.

CFM—Air flow rate, cubic feet per minute.

T_1, T_2, T_3—The water temperature as indicated in Fig. 5.

t_1, t_2, t_3—The air temperatures as indicated in Fig. 5.

W.B. or Wet Bulb—The temperature at which water, by evaporating into air, can bring the air to saturation at the same temperature.

h_{a1}, h_{a2}, h_{a3}—The enthalpies which correspond to the air wet bulb temperatures at points 1, 2 and 3.

Simpson's Rule for areas—Simpson's rule states, "Divide the area into n panels (where n is some even num-

Fig. 2—The air-out temperature of a tower is a function of the air-in wet bulb temperature, the water rate and the air rate.

gpm × min/hr × lb/gal = lb H_2O/ hr.

$\Delta h = 111$ gpm × Δt_w/cfm

$\Delta h + h_{a3} = f\ (t_{a1})$

The air-out enthalpy, h_{a1}, is equal to the change of enthalpy which the air undergoes while within the tower plus the enthalpy of the entering air.

Fig. 2

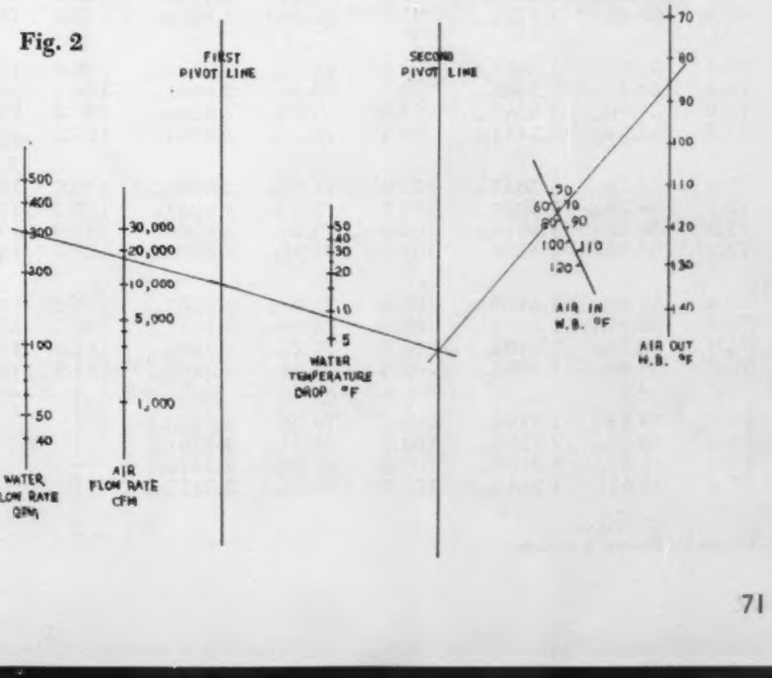

TABLE 1. SPECIFIC ENTHALPY & COOLING POTENTIAL[1]

Temp. F	Spec Enth, Btu/lb dry air *	Cooling Pot, in. Hg *	Temp. F	Spec Enth, Btu/lb dry air *	Cooling Pot, in. Hg *	Temp. F	Spec Enth, Btu/lb dry air *	Cooling Pot, in. Hg *	Temp. F	Spec Enth, Btu/lb dry air *	Cooling Pot, in. Hg *
50.0	20.28_{28}	0.8988_{122}	78.0	41.52_{52}	1.8024_{213}	106.0	83.29_{106}	3.4238_{388}	132.0	164.40_{226}	6.0956_{670}
50.5	20.56_{28}	0.9110_{123}	78.5	42.04_{52}	1.8237_{215}	106.5	84.35_{107}	3.4626_{391}	132.5	166.66_{230}	6.1626_{676}
51.0	20.84_{28}	0.9233_{125}	79.0	42.56_{53}	1.8452_{217}	107.0	85.42_{109}	3.5017_{396}	133.0	168.96_{234}	6.2302_{683}
51.5	21.12_{29}	0.9358_{125}	79.5	43.09_{54}	1.8669_{220}	107.5	86.51_{110}	3.5413_{400}	133.5	171.20_{237}	6.2985_{690}
52.0	21.41_{29}	0.9483_{126}	80.0	43.63_{54}	1.8889_{222}	108.0	87.61_{113}	3.5813_{404}	134.0	173.67_{241}	6.3675_{697}
52.5	21.70_{29}	0.9609_{128}	80.5	44.17_{55}	1.9111_{224}	108.5	88.74_{114}	3.6217_{409}	134.5	176.08_{244}	6.4372_{705}
53.0	21.99_{30}	0.9737_{130}	81.0	44.72_{55}	1.9335_{227}	109.0	89.87_{115}	3.6626_{413}	135.0	178.52_{248}	6.5077_{712}
53.5	22.29_{30}	0.9867_{130}	81.5	45.27_{56}	1.9562_{229}	109.5	91.02_{117}	3.7039_{417}	135.5	181.00_{251}	6.5789_{720}
54.0	22.59_{30}	0.9997_{131}	82.0	45.83_{57}	1.9791_{231}	110.0	92.19_{119}	3.7456_{422}	136.0	183.51_{255}	6.6509_{728}
54.5	22.89_{31}	1.0128_{133}	82.5	46.40_{58}	2.0022_{234}	110.5	93.38_{120}	3.7878_{426}	136.5	186.06_{260}	6.7237_{734}
55.0	23.20_{31}	1.0261_{134}	83.0	46.98_{58}	2.0256_{236}	111.0	94.58_{122}	3.8304_{431}	137.0	188.66_{264}	6.7971_{741}
55.5	23.51_{31}	1.0395_{135}	83.5	47.56_{60}	2.0492_{238}	111.5	95.80_{123}	3.8735_{435}	137.5	191.30_{270}	6.8712_{749}
56.0	23.82_{31}	1.0530_{136}	84.0	48.16_{60}	2.0731_{242}	112.0	97.03_{125}	3.9170_{440}	138.0	194.00_{274}	6.9461_{757}
56.5	24.13_{32}	1.0666_{138}	84.5	48.76_{60}	2.0973_{244}	112.5	98.28_{127}	3.9610_{445}	138.5	196.74_{280}	7.0218_{765}
57.0	24.45_{33}	1.0804_{139}	85.0	49.36_{62}	2.1217_{247}	113.0	99.55_{129}	4.0055_{449}	139.0	199.54_{285}	7.0983_{772}
57.5	24.78_{32}	1.0943_{140}	85.5	49.98_{62}	2.1464_{249}	113.5	100.84_{130}	4.0504_{455}	139.5	202.39_{289}	7.1755_{780}
58.0	25.10_{33}	1.1083_{142}	86.0	50.60_{63}	2.1713_{252}	114.0	102.14_{132}	4.0959_{460}	140.0	205.28_{294}	7.2535_{788}
58.5	25.43_{33}	1.1225_{143}	86.5	51.23_{63}	2.1965_{255}	114.5	103.46_{134}	4.1419_{464}	140.5	208.22_{299}	7.3323_{796}
59.0	25.76_{33}	1.1368_{145}	87.0	51.86_{65}	2.2220_{257}	115.0	104.80_{136}	4.1883_{469}	141.0	211.21_{304}	7.4119_{804}
59.5	26.09_{34}	1.1513_{146}	87.5	52.51_{65}	2.2477_{260}	115.5	106.16_{138}	4.2352_{474}	141.5	214.25_{310}	7.4923_{813}
60.0	26.43_{34}	1.1659_{147}	88.0	53.16_{66}	2.2737_{263}	116.0	107.54_{140}	4.2826_{479}	142.0	217.35_{315}	7.5736_{821}
60.5	26.77_{35}	1.1806_{149}	88.5	53.82_{67}	2.3000_{266}	116.5	108.94_{142}	4.3305_{484}	142.5	220.50_{320}	7.6557_{829}
61.0	27.12_{35}	1.1955_{150}	89.0	54.49_{68}	2.3266_{269}	117.0	110.36_{144}	4.3789_{490}	143.0	223.70_{325}	7.7386_{837}
61.5	27.47_{35}	1.2105_{152}	89.5	55.17_{68}	2.3535_{271}	117.5	111.80_{147}	4.4279_{494}	143.5	226.95_{331}	7.8223_{846}
62.0	27.82_{36}	1.2257_{154}	90.0	55.85_{70}	2.3806_{275}	118.0	113.27_{149}	4.4773_{500}	144.0	230.26_{336}	7.9069_{855}
62.5	28.19_{36}	1.2411_{155}	90.5	56.55_{70}	2.4081_{277}	118.5	114.76_{150}	4.5273_{505}	144.5	233.62_{341}	7.9924_{863}
63.0	28.54_{37}	1.2566_{156}	91.0	57.25_{72}	2.4358_{282}	119.0	116.26_{153}	4.5778_{511}	145.0	237.03_{346}	8.0787_{872}
63.5	28.91_{37}	1.2722_{158}	91.5	57.97_{72}	2.4640_{283}	119.5	117.79_{155}	4.6289_{516}	145.5	240.49_{353}	8.1659_{880}
64.0	29.28_{37}	1.2880_{160}	92.0	58.69_{73}	2.4923_{287}	120.0	119.34_{157}	4.6805_{521}	146.0	244.02_{359}	8.2539_{888}
64.5	29.65_{38}	1.3040_{161}	92.5	59.42_{74}	2.5210_{289}	120.5	120.91_{160}	4.7326_{527}	146.5	247.61_{366}	8.3427_{898}
65.0	30.03_{38}	1.3201_{163}	93.0	60.16_{75}	2.5499_{293}	121.0	122.51_{163}	4.7853_{533}	147.0	251.27_{374}	8.4325_{908}
65.5	30.41_{39}	1.3364_{164}	93.5	60.91_{76}	2.5792_{296}	121.5	124.14_{164}	4.8386_{538}	147.5	255.01_{381}	8.5233_{917}
66.0	30.80_{39}	1.3528_{166}	94.0	61.67_{77}	2.6088_{299}	122.0	125.78_{167}	4.8924_{544}	148.0	258.82_{389}	8.6150_{925}
66.5	31.19_{39}	1.3694_{168}	94.5	62.44_{78}	2.6387_{300}	122.5	127.45_{170}	4.9468_{549}	148.5	262.71_{396}	8.7075_{935}
67.0	31.58_{40}	1.3862_{170}	95.0	63.22_{79}	2.6690_{305}	123.0	129.15_{173}	5.0017_{556}	149.0	266.67_{404}	8.8010_{944}
67.5	31.98_{40}	1.4032_{171}	95.5	64.01_{80}	2.6995_{310}	123.5	130.88_{174}	5.0573_{561}	149.5	270.71_{411}	8.8954_{953}
68.0	32.38_{41}	1.4203_{173}	96.0	64.81_{81}	2.7305_{313}	124.0	132.62_{177}	5.1134_{567}	150.0	274.82_{419}	8.9907_{963}
68.5	32.79_{41}	1.4376_{174}	96.5	65.62_{82}	2.7618_{315}	124.5	134.39_{180}	5.1701_{573}	150.5	279.01_{427}	9.0870_{973}
69.0	33.20_{42}	1.4550_{177}	97.0	66.44_{84}	2.7933_{319}	125.0	136.19_{183}	5.2274_{579}	151.0	283.28_{435}	9.1843_{981}
69.5	33.62_{42}	1.4727_{178}	97.5	67.28_{84}	2.8252_{322}	125.5	138.02_{185}	5.2853_{585}	151.5	287.63_{442}	9.2824_{992}
70.0	34.04_{43}	1.4905_{180}	98.0	68.12_{86}	2.8574_{327}	126.0	139.87_{188}	5.3438_{591}	152.0	292.05_{450}	9.3816_{1002}
70.5	34.47_{43}	1.5085_{182}	98.5	68.98_{87}	2.8901_{329}	126.5	141.75_{191}	5.4029_{598}	152.5	296.55_{458}	9.4818_{1012}
71.0	34.90_{44}	1.5267_{184}	99.0	69.85_{88}	2.9230_{333}	127.0	143.66_{194}	5.4627_{605}	153.0	301.13_{466}	9.5830_{1021}
71.5	35.34_{44}	1.5451_{186}	99.5	70.73_{89}	2.9563_{337}	127.5	145.60_{196}	5.5232_{610}	153.5	305.79_{475}	9.6851_{1032}
72.0	35.78_{45}	1.5637_{187}	100.0	71.62_{90}	2.9900_{341}	128.0	147.56_{199}	5.5842_{616}	154.0	310.54_{484}	9.7883_{1042}
72.5	36.23_{45}	1.5824_{190}	100.5	72.52_{92}	3.0241_{344}	128.5	149.55_{202}	5.6458_{623}	154.5	315.38_{494}	9.8925_{1052}
73.0	36.68_{46}	1.6014_{192}	101.0	73.44_{93}	3.0585_{348}	129.0	151.57_{205}	5.7081_{630}	155.0	320.32_{505}	9.9977_{1061}
73.5	37.14_{46}	1.6206_{194}	101.5	74.37_{94}	3.0933_{353}	129.5	153.62_{208}	5.7711_{635}	155.5	325.37_{516}	10.1038_{1072}
74.0	37.60_{47}	1.6400_{195}	102.0	75.31_{95}	3.1285_{356}	130.0	155.70_{212}	5.8346_{642}	156.0	330.53_{526}	10.2110_{1084}
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76.5	40.01_{50}	1.7399_{206}	104.5	80.21_{101}	3.3101_{375}				158.5	357.96_{582}	10.7634_{1137}
77.0	40.51_{50}	1.7605_{208}	105.0	81.22_{103}	3.3476_{379}				159.0	363.78_{594}	10.8771_{1149}
77.5	41.01_{51}	1.7813_{211}	105.5	82.25_{104}	3.3855_{383}				159.5	369.72_{606}	10.9920_{1159}
									160.0	375.78	11.1079

[1]From F. Rasori's article.

ber) by means of n + 1 parallel lines, called ordinates, drawn at constant distance h apart; denote the lengths of these ordinates as $y_o, y_1, y_2, y_3, y_4, \ldots, y_n$. The equation for the total area is —

$$A = 1/3\ h\ (y_o + 4y_1 + 2y_2 + 4y_3 + 2y_4 + \ldots + y_n)$$

A = the total area
h = the horizontal width divided by n
n = the number of panels comprising the area

Divide Fig. 5 into two sub areas or panels and label the diagram as below—

The main point of interest is not the total area, but rather, the average or "mean" distance between the upper and lower curves. This difference is a measure of the cooling potential difference. Thus, first derive the form of the Simpson equation to fit the area of Fig. 6:

$$A = 1/3 \frac{(b - a)}{n} (y_o + 4y_1 + y_n)$$

Next, divide both sides of the equation by (b — a) in order to convert the area to the average or "mean" height. This gives the final form—

$$\frac{A}{(b - a)} = \frac{1/3}{n} (y_o + 4y_1 + y_n) =$$

mean cooling potential difference or height, where n is equal to 2 in this case.

A discussion of Simpson's rule may be found in Marks' *Mechanical Engineers' Handbook*, Fifth Edition, p 104.

Nomograph construction—It is possible to construct a nomograph for very complex equations but the basic procedure is one of mechanical drawing for all types of nomographs. Below is the nomograph for a simple equation, a + b = c. This graph was created by taking several definitely formalized steps which are basic. First, it is necessary to determine the range of each scale; this is determined by the type of problem to be solved and by the numerical range of the solution. The scales for the left hand side of the above equation are laid off and the equation is ready for solution. It is then necessary to draw some intercepting trial solution lines for the equation. For example:

1. Set off the outside scales as below—
2. Draw some trial lines to determine at least three points on the answer scale—
3. Draw in the answer scale—
4. Determine the position of all other major points on the answer scale—

Step 4. results in the final nomograph for the equation, a + b = c. If the reader is interested in more complex charts he may refer to: M. G. Van Voorhis, "How to Make Alignment Charts" McGraw-Hill Book Co., Inc., New York, 1937.

Fig. 3

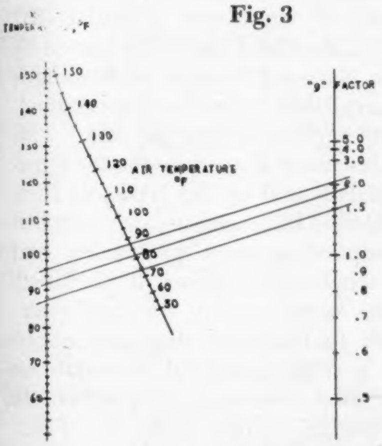


Fig. 3—the "g" factor scale is a simplified version of the water-air cooling potential difference. Table 1 lists the cooling potential at a total pressure of 29.9212 inches mercury. Table 1 can be used in lieu of Fig. 3; simply determine the reciprocal of the difference between the water and air cooling potentials.

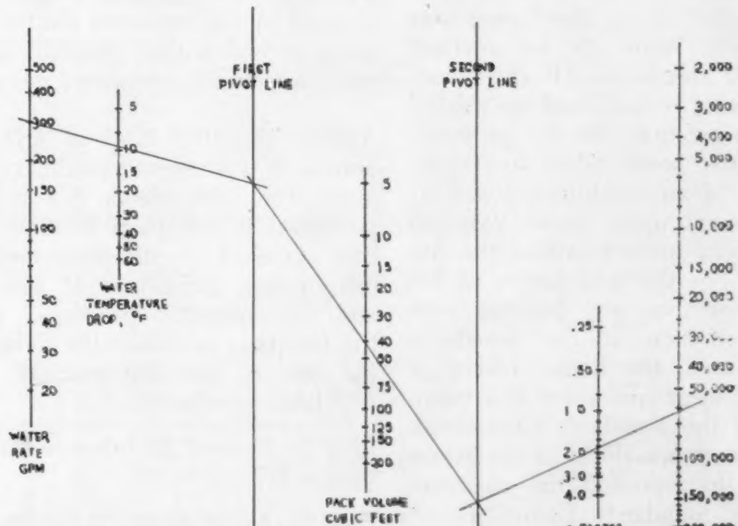
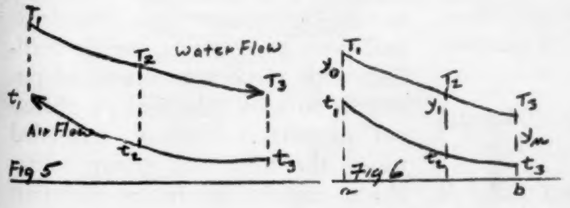


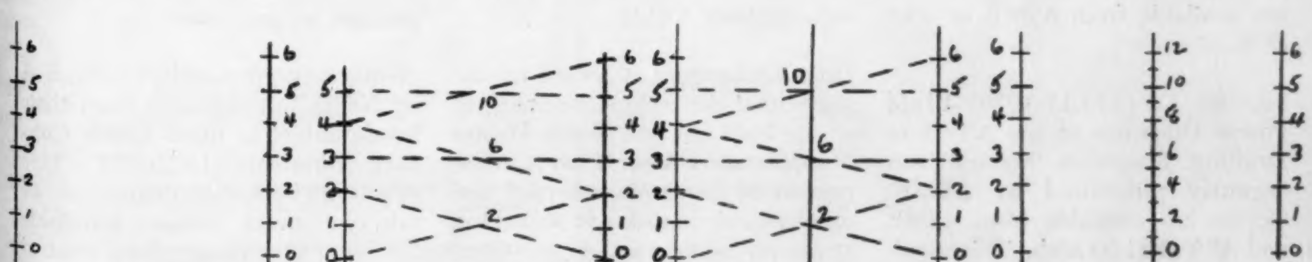
Fig. 4

Fig. 4—The "G" factor noted here is the summation of the reciprocals of the cooling potentials. This diagram represents the equation:

$$Ka = \frac{500 \times gpm \times \Delta Tw}{\text{pack volume}} \times \left[\frac{1}{\theta w - \theta a}\right]_m$$

$$\text{where } \left[\frac{1}{\theta w - \theta a}\right]_m = \sum_{1:4:1} \left[\frac{1}{\theta w - \theta a}\right]$$





All-society responsibility

A. T. BOGGS III

ASHRAE Technical Secretary

Active society standards—The project committees of your Standards Committee presently active in the development of new standards and revisions to existing standards are listed on the facing page. Of the 19 committees listed, it will be noted that 13 are developing new standards while six are revising existing standards. Of the standards being revised, half are related to refrigeration. Of the proposed standards, seven relate to specific phases of air conditioning and six to general areas allied with all phases of interest within the Society. For the information of Society members not familiar with the operation of the Standards Committee, the liaison officer of each project committee is a member of the Standards Committee. It is the responsibility of the liaison officer to represent the chairman of the Standards Committee at meetings of the project committees and assist these committees in complying with the policies established by the Standards Committee.

ASA-Y14: Two additional sections of the American Drafting Standards Manual were recently announced by ASME. Section 10 (Y14.10-1959) covers **Metal Stampings.** The purpose of this section is to indicate basic empirical layout and drafting practices specifically related to metal stamping. Copies are available from ASME or ASA at $1.50 each.

Section 17 (Y14.17-1959) **Fluid Power Diagrams** of the American Drafting Standards Manual was recently published by ASME. Copies are available from ASME and ASA at $1.50 each. This standard includes procedures to be followed in the drawings which depict fluid power systems. Systems covered by this standard are those using a fluid within enclosed circuits to transmit and control power.

ASTM: The 1958 Book of ASTM Standards was recently made available. The 1958 edition has been published in ten parts containing 2450 standard specifications, methods of test, definitions of terms, and recommended practices. Of the ten parts available, the following are of primary interest to ASHRAE members:

Part 1 – Ferrous Metals Specifications – $12

Part 2 – Non-Ferrous Metals Specifications, Electronic Materials – $10

Part 3 – Metals Test Methods–$10

Part 4 – Cement, Concrete, Mortars, etc. – $12

Part 5 – Machinery Products, Thermal Insulation, Building Construction, etc. – $12

Part 9 – Plastics Electrical Insulation – $14

The complete set of ten parts is available at $116. Copies may be ordered from ASTM.

Heat Exchangers: A recent release stated that the technical committee of the **Fuel Oil and Water Heater Manufacturers Association** is in the process of developing thermal and mechanical standards covering steam to water, water to water, and other types of shell and tube heat exchangers. When complete, these standards will be available in brochure form to engineers and architects for specifying purposes. The association's address is: 50 East 42 Street, New York 17, N. Y.

Standard materials: National Bureau of Standards Circular 552 titled **Standard Materials Issued by the National Bureau of Standards** is available from the Superintendent of Documents at 35c. This publication lists the standard materials issued by the NBS and their prices. Information on certified composition and properties and purchase procedures is included. NBS began issuing standard materials in 1905 with the preparation of a small group of materials on certified chemical composition. In response to the demands of science and industry, this program has grown and now NBS supplies more than 550 different standards of metals, ores, ceramics, chemicals, hydrocarbons, and radioactive materials. When these materials were first issued they were known as Standard Samples since they were used as standards in chemical analysis. This term was extended to similar composition standards and later to cover materials certified with respect to chemical purity or to some physical or chemical property. Through extended usage the term has grown to include certain reference materials issued without certification of composition or properties.

Nema: A recent standard published by Nema and available from their headquarters is titled **Quick Connect Terminals, DC2-1959.** This standard indicates definitions of tab type quick connect terminals for automatic temperature control devices. Copies are available at no charge.

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J. R. Caulk, Jr.
A. S. Decker
W. L. McGrath
F. J. Reed
J. A. Schenk
S. P. Soling
J. D. Loveley
J. A. Galazzi
J. L. Ditzler
K. T. Davis
A. Giannini
F. Janssen
G. B. Priester
H. Wolf

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BUILDING
P. J. Marschall, *Chairman*
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D. H. McCuaig, *Chairman*
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M. C. Turpin

ASHRAE-AIA COOPERATION
J. E. Haines, *Chairman*
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A. J. Hess
H. C. Hoffmann
G. A. Linskie
R. J. Salinger
J. L. Wolf
P. H. Yeomans

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Staff and Administration
C. Boling
E. R. Queer
P. B. Gordon

Finance
J. H. Fox
F. Y. Carter
C. Boling
E. R. Queer

Non-Technical and Regional
A. J. Hess
W. A. Grant
D. D. Wile

Technical
H. F. Spoehrer
R. H. Tull
John Everetts, Jr.

UEC FUND RAISING
M. F. Blankin, *Chairman*
J. E. Haines, *Vice Chairman*
C. F. Holske

Others are saying—

that . . . . . . Heating outdoor air for a nickel mine by the recovery heat exchange system has proved successful in underground mining, and also has possibilities for use in other underground areas. Air exhausted from the mine is passed over a large coil through which water is circulated. Heated water then goes through another coil which heats incoming outdoor air. *Heating, Piping and Air Conditioning, May 1959, p 114.*

that . . . . . . results reported recently from Sweden and Belgium on heat losses from underground mains, discussed and compared with calculations by methods proposed by the author in an earlier publication, are the subject of this presentation. *Journal of the Institution of Heating and Ventilating Engineers, April 1959, p 22.*

that . . . . . . sufficient price advantage exists between the cost per million Btu's for a coal-fired plant as compared to oil-fired plants to amortize both the capital investment differential and interest charges within a relatively short number of years. Many exorbitant claims are made about savings in labor cost that can be affected by converting from coal to oil. A careful study at state-owned boiler plants will show that little, if any, labor can be saved. *Coal Heat and Building Materials, April 1959, p 10.*

that . . . . . . the only means of preventing the contamination of gases during compression, in the range of working conditions in which turbocompressors cannot be used, is to employ piston compressors so designed that no normal or substitute lubricants come into contact with the medium handled. *Sulzer Technical Review, 1958/2 p 3.*

that . . . . . . present knowledge of the mechanism of nucleate boiling is slight, especially regarding the conditions necessary for bubble formation. In research carried on to investigate some of the factors which influence nucleate boiling, an attempt was made to find a correlation of the conditions necessary to form bubbles. As the pressure increases, the amount of superheat required to start ebullition, and to support nucleate boiling, decreases. The tests made with varying tube diameters and surface finish, showed that the diameter has little effect on the region of natural convection, or on the point where ebullition begins, but like pressure, its effect in the region of nucleate boiling is quite considerable. It was found that the surface finish has a great effect on the amount of superheat required to start ebullition. *Journal of Refrigeration, March/April 1959, p 32.*

that . . . . . . it is a matter of some importance for an air conditioning engineer to know what coil proportions will be required to accomplish a given cooling and dehumidification range. Attention is focused mainly on contraflow coils having chilled water or brine as the cooling medium, in this paper on a method of predicting cooling coil performance. Step-by-step procedures are established for everyday use in determining the size of coil required for a specific duty. Contraflow direct expansion coils are also covered. *Journal of the Institution of Heating and Ventilating Engineers, April 1959, p 1. (British).*

that . . . . . . vertical sunshades of solar-reducing glass have been a design feature of several buildings recently. Advantage of these sunshades are reduction of heat and glare without moving parts or hiding the view.

Meetings ahead

June 1-4 – National District Heating Association, 50th Anniversary, Skytop, Pa.

June 4-5–National Warm Air Heating and Air Conditioning Association, semi-annual meeting, Statler-Hilton Hotel, Cleveland, Ohio.

June 8-9 – A Short Course on Industrial Water Conservation, University of Michigan, School of Public Health, Ann Arbor, Mich.

June 11-13–Heat Transfer and Fluid Mechanics Institute, University of California, Los Angeles, Calif.

June 14-18–American Society of Mechanical Engineers, semi annual Meeting, Chase-Park Plaza Hotel, St. Louis, Mo.

June 14-19 – Electrical Precipitation Seminar, Pennsylvania State University, University Park, Pa.

June 17-20–National Society of Professional Engineers, Engineering Progress Exposition, Hotel Commodore, New York, N. Y.

June 22-24–American Society of Heating, Refrigerating and Air-Conditioning Engineers, annual meeting, Lake Placid, N. Y.

August 19-26–10th International Congress of Refrigeration, Copenhagen, Denmark.

September 2-4–Cryogenic Engineering Conference, University of California, Berkeley, Calif.

September 25-29–American Meat Institute, Palmer House, Chicago, Ill.

October 5-7–American Gas Association, Annual Convention, Chicago, Ill.

October 30-November 2 – Refrigeration Service Engineers Society, Annual Convention, Atlantic City, N. J.

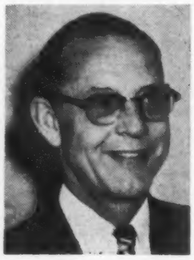
November 2-5–11th Exposition of the Air-Conditioning and Refrigeration Industry, Atlantic City, N. J.

February 1-4 – American Society of Heating, Refrigerating and Air-Conditioning Engineers, semi annual Meeting, Dallas, Texas.

February 1-4–2nd Southwest Heating and Air-Conditioning Exposition, Dallas, Texas.

People

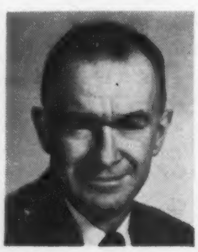
Aubrey I. Brown, Professor of Mechanical Engineering, Ohio State University, died recently. A life member of the former ASHAE, he contributed to several editions of the Heating, Ventilating and Air Conditioning Guide, and served on the Publication Committee and on the Technical Advisory Committee on Heating Load. He was co-author of the textbook, "Introduction to Heat Transfer." Professor Brown devoted more than forty years to engineering teaching, and served as chairman of the Department of Mechanical Engineering at Ohio State from 1946 to 1952. At that time he resigned from administrative duties in order to give his full time to teaching. He also served as consultant on heating, ventilating and air conditioning for various Ohio concerns, state and county offices, and for the Association of American Railroads.



Ralph M. Westcott, Consulting Engineer, Holladay and Westcott, has been named President of the Consulting Engineers Council, a national organization of 24 groups of owner-principals active in consulting work and affiliated with Engineers Joint Council. President Westcott took office at the annual meeting in New York for a one-year term. A member of ASRE from 1943, he was a member of Council (1946-1957) and served on several committees prior to the formation of ASHRAE.

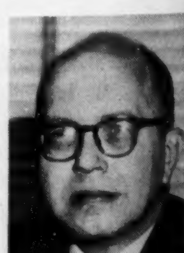
Edward Von Arb was recently named Vice President in charge of engineering for Revco, Inc. He has been with the firm since 1956 as Director of Engineering. Previously, he served as Refrigeration Product Manager for the Whirlpool Corporation, and had been with Servel, Inc., for 19 years before that in various engineering capacities having to do with household and commercial refrigeration. A graduate of the University of Louisville, BME, he is a registered professional engineer in the state of Indiana.

William Fay has been appointed to the newly created post of Chief Design Engineer with Minneapolis-Honeywell Regulator Company, in the Temperature Controls Group. In this post, he will supervise the design of several products. He has been with the firm since 1945 and has served in the capacities of Test Engineer, Field Engineer, Assistant to the Chief Mechanical Engineer, Supervisor and as Assistant Market Manager for residential air conditioning. His work has been largely in the design and applications development field.



John R. Jamieson steps into the position of Chief Field Engineer in the Temperature Controls Group of Minneapolis-Honeywell Regulator Company. He holds an MS degree in Mechanical Engineering from the University of Minnesota, and before joining his present firm in 1952, was an instructor and Research Associate at the University of Illinois. He is the author of several papers having to do with perimeter heating. During the last three years, he has held the post of Application Engineer for the residential air conditioning market.

Hans P. Peterson becomes Chief Engineer, Recony Corporation, manufacturers of ground support equipment for testing purposes. A graduate engineer of Worcester Polytechnic Institute (Chemical Engineering), he was formerly Chief Engineer, Dunham-Bush, Inc., and most recently, with Kramer Trenton Company as district representative for New York. He has been active in the field of aircraft air conditioning and has done original design and development work upon essential components for this field.



Alwin B. Newton, Director of Engineering for the York Div of Borg-Warner Corporation, has also been named a Vice President of the company. With degrees in Mechanical Engineering from Syracuse University and Massachusetts Institute of Technology, he began his career as a student engineer for General Electric Company. He has since held key posts with Minneapolis-Honeywell Regulator Company, Chrysler Corporation and others. Immediately before joining his present firm, Mr. Newton served as Vice President of research and design for the Coleman Company. He is the holder of nearly 200 patents in the fields of heating, air conditioning and related areas.

Rudolph G. Berg, Vice President of Copeland Refrigeration Corporation, was elected president of the Air-Conditioning and Refrigeration Institute at its annual meeting in May. Vice President for the past year, Mr. Berg has been chairman of the association's Tax Committee for the past three years. Other officers newly-elected are: Vice President **Russell Gray,** Vice President of Carrier Corporation; Vice President **L. N. Hunter,** Vice President of National-U. S. Radiator Corporation; Vice President **R. K. Serfass,** General Manager of the Air Conditioning Div, Westinghouse Electric Corporation; and Treasurer **W. H. Aubrey,** Vice President of Frick Company. Mr. Berg succeeds **Don V. Petrone,** President of Typhoon Air Conditioning Company, as head of the Institute.

David T. Donovan, former Assistant Secretary of ASHAE, moves from his position as Manual and Data Specialist in the Home Heating and Cooling Department of General Electric Company to the role of sales planner with the same department. He will continue his work in Tyler, Texas.

William F. Ryan will receive the 1959 National Society of Professional Engineers award for outstanding service to the engineering profession, at their meeting this month. His citation reads, in part, for his "outstanding contributions as a professional engineer in the field of mechanical design . . . and . . . his untiring efforts in the practical implementation of these concepts for the benefit of the profession and mankind." He is a former Vice President of Stone and Webster Engineering Corporation.

Walter L. Fleisher, President of Air and Refrigeration Corporation of New York and Atlanta, died recently as the result of injuries received in an automobile accident. A life member and former President of ASHAE, he received the F. Paul Anderson award in 1953, and was also an editor of the Heating, Ventilating and Air Conditioning Guide and a member of the Advisory Council to the Department of Mechanical Engineering at Princeton University.

He was graduated from the University of Pennsylvania, and was presented an honorary degree in Mechanical Engineering from that school in 1931. In 1955, he received a citation for his work from the engineering schools of the university.

As Engineer-in-Chief for the firm of Francis Bros. & Jellett in 1903-1911, he was one of the first to design and install air conditioning equipment in the United States. In 1911, Mr. Fleisher formed W. L. Fleisher & Company, Inc., serving as President, Treasurer and Chief Engineer until its merger in 1925 with two other companies to form the Cooling and Air Conditioning Corporation. He remained as Vice President and Chief

Engineer until 1930. That year he established a consulting firm and four years later became President of the Air and Refrigeration Corporation.

He was the holder of some hundred patents covering practically all phases of cooling, ventilating, controlling, cleaning, drying and processing. His patents on local recirculation and the Fleisher by-pass method were basic to all comfort air conditioning.

BULLETINS

Automatic Boilers. Data on packaged automatic boilers in sizes from 20 through 100 hp are given in this 8-page bulletin on a new line of mechanical pressure atomizing oil-fired and partial pre-mixing gas-fired units. Illustrated and explained, these systems are guaranteed at 80% efficiency, and have high pressure designs to 250 psi, a special hot water design, electronic Fireye combustion control with prepurge and postpurge cycles, and low fire start with optional high/low firing on 40 hp and larger.
Orr & Sembower, Inc., Morgantown Rd., Reading, Pa.

Fan Components. Interchangeable-hub fan blades, in the improved Adapter series, have uniform size openings and are in six sizes from 10 to 24 in. as described in this flyer.
Ventco Inc., 2900 Caroline St., Houston 4, Texas.

Air-Cooled Condensers. For those designing and specifying equipment for the high side of air conditioning or refrigeration plants, this manufacturer offers a two-page bulletin on his air-cooled condenser product. It includes details on multiple circuit application data and application information on condensing pressure controls. Bulletin ACC, 5-1-58.
Drayer-Hanson, P. O. Box 2215, Terminal Annex, Los Angeles 54, Calif.

Filtered Sterile Water. Production of sterile pyrogen-free water from boiler steam is described in this 8-page bulletin containing schematic diagrams, specifications and a detailed discussion of the major features of the system. It operates on the basic principle that all contaminants in steam are entrained in particles of moisture.

Operation of the Steril-Aqua system, ranging from 5 to 500 gph, is explained in the Bulletin, No. IN-201.
Wilmot Castle Company, 1934 E. Henrietta Rd., Rochester, N. Y.

Oscillogram Processor. For on-the-spot processing, this self-contained motorized Oscillogram Processor is designed for daylight processing of paper records. Features described in Bulletin 1537 include a thermistor drum temperature control, variable bath temperature, a speed-control contact shoe making it possible to

(Continued on page 84)

What ASHRAE Chapters are doing

Controls, equipment and installations pertaining to air conditioning were the most discussed topics at late spring meetings. Heat pumps, testing facilities and allied interests also received attention.

ROCKY MOUNTAIN (H) . . . Electromechanical heating and cooling elements which can heat or cool depending upon the direction of current flow, or when heated and cooled simultaneously at opposite ends can produce electrical current flow, namely, the Seebeck Effect, was the subject of Robert S. Lackey at the April meeting.

Mr. Lackey, with Westinghouse Electric Corporation in the New Products Engineering Dept, explained how refrigerators, space heaters or power generators may be made with no moving parts. Mr. Lackey similarly addressed the 45th semi annual meeting of the former ASRE in New Orleans.

BIRMINGHAM (R) . . . At the early May meeting of this chapter, members were to hear John Spence on Recurrent Trends in Food Store Refrigeration Equipment. Mr. Spence is the Senior Manager of Hussmann Refrigeration Company.

DETROIT (R) . . . Daniel Kramer, Kramer-Trenton Company, who addressed the former ASRE 54th Annual Meeting on Refrigerant Control in Air-Cooled Condensers, planned to meet with these members in May to discuss problems in system operation which arise during winter operation. He then evaluated the mechanisms which are now used to solve these problems.

PITTSBURGH . . . Polyvinyl chloride, used in Germany and in marine equipment during the world war, was introduced into America in 1946 and in three years a demand for the plastics product was created. The development of the market for plastics materials was the subject of a talk by Kenneth Chamberlain, speaking on the Use of Plastics Materials in Heating and Air Conditioning.

Plastics duct work, fans and piping for handling corrosive fumes are important in the air conditioning and heating field, he noted. Naval ships are being equipped with 100% PVC piping for anti-radiation contamination sprays, he pointed out. Mr. Chamberlain went on to explain that although PVC piping and fittings run twice the cost of steel, recent surveys have indicated that total savings may run as high as 30% because of installation speed.

The speaker displayed various types of plastics valves, springs, nuts, bolts and fittings. He introduced a new plastics material from Germany, with the same coefficient of expansion as steel. A chemical company has arranged to manufacture the basic resin, named Bascodur, he told members. There was a lengthy question and answer period.

CHICAGO (R) . . . Inspecting the Bell & Gossett Company plant in May, these members viewed the line of products from experimental engineering through machining, assembly and testing.

KANSAS CITY (H) . . . At the last meeting of the season, members invited Cary B. Gamble, consulting engineer, to speak on Air Conditioning Systems.

WESTERN MICHIGAN (R) . . . Touring the refrigerator manufacturing plant of the Norge Div of Borg-Warner Corporation, members meeting here at the beginning of April viewed the automatic welding lines for both cabinet exteriors and interiors in operation. They were addressed by H. W. Whitmore, Director of Engineering.

CONNECTICUT VALLEY (R) . . . Preceding the regular May meeting, there was to be a tour of the Chandler-Evans Corporation, Div of Pratt & Whitney Small Tool Company, featuring the inspection of high and low temperature test facilities for fuel systems. After dinner, E. J. Vitali, Manager

CHAPTER AND SECTION DATES*

	June
Atlanta (H)	—
Austin (H)	—
Arizona (H)	1
Arkansas (H)	—
Baltimore (H)	—
Baltimore (R)	—
Baton Rouge (H)	—
Birmingham (R)	11
Bluegrass (H)	—
British Columbia (H)	—
Boston (R)	—
Buffalo (R)	4
Carolina (R)	—
Central Michigan (R)	18
Central New York (H)	—

* Pending establishment of unified ASHRAE chapters, meetings of former ASHAE Chapters (H) and former ASRE Sections (R) are proposed upon the basis indicated above for the month of June.

	June
Central New York (R)	11
Central Ohio (H)	—
Central Pennsylvania (R)	12
Chicago (R)	—
Cincinnati (H)	—
Cincinnati	—
Cleveland (R)	—
Colorado (R)	—
Columbus (R)	—
Connecticut (H)	18
Connecticut Valley (R)	5
Dallas-Fort Worth (R)	—
Dayton (R)	—
Delta (H)	—
Detroit (R)	—
El Paso (H)	—
Empire State Capital (H)	—
Erie (R)	9
Evansville (R)	5
Florida West Coast (R)	10
Fort Worth (H)	—

	June
Georgia (R)	8
Golden Gate (H)	—
Hudson-Mohawk (R)	—
Houston (R)	—
Illinois (H)	—
Illinois-Iowa (H)	—
Indiana (H)	—
Indianapolis (R)	20
Inland Empire (H)	—
Iowa (H)	—
Jacksonville (H)	2
Johnstown (H)	—
Kansas (H)	—
Kansas City (H)	—
Kansas City (R)	15
Long Island (H)	—
Los Angeles (R)	15
Louisville (R)	—
Manitoba (H)	—
Massachusetts (H)	—
Memphis (H)	15

of Engineering, Test and Experimental Dept, Chandler-Evans Corporation, chose as his topic, Effect of Temperatures in Chandler-Evans Product Development.

REGION 2 . . . ASHRAE President Cecil Boling addressed the conference in April of eight Canadian Chapters in a full day devoted to matters of internal

Addressing the Region 2 meeting on post merger developments is ASHRAE President Cecil Boling. At the head table with him are Daniel D. Wile, 2nd Vice President; W. G. Hole, Regional Director, Region 2; and Walter A. Grant, 3rd Vice President. Directly across from Mr. Boling is Hayward Murray, first President of the merged Montreal Chapter.

organization. The speaker at both the luncheon and the coffee talk, he noted that the merging of the Societies and the activities of the Societies has been taking place at a more rapid pace than was originally anticipated.

In the evening, Dr. Victorien Fredette, Associate Professor of Bacteriology, University of Montreal, presented his views on the Bacteriological Problems in Ventilating Systems.

Simultaneously, with the Region 2 conference in Montreal, the executive committee of ASHRAE met in the same city. Plans for future policies and activities were discussed, and preparations for the Lake Placid meeting in June approved.

RHODE ISLAND (H) . . . How electric heating may be used in residential and commercial installation was explained by Ted Lanan, Electric Heating Engineer, Edwin L. Wiegand Company. He showed slides of various electric heating installations throughout the country and compared the cost of electric heating with fossil fuels.

Don Kenny, also of the Wiegand Company, discussed the various types of electric heating equipment on display.

MILWAUKEE (R) . . . We are merely scratching the surface of applications for the heat pump, said R. G. Werden, pioneer with the compound heat pump, in his talk before this chapter in April. Supplementing his lecture with slides, he defined the heat pump and discussed actual installation. Throughout his presentation, the speaker stressed that the application of the heat pump is based upon the basic concept of conservation of energy. Btus normally wasted are put to practical use in this unit, he emphasized.

NORTHERN OHIO (H) . . . Describing the air conditioning system at the Cleveland Press Building, Wesley Hunting, Austin Company, explained the problems and their solutions which were peculiar to this job, as control of relative humidity and special filter arrangements used to subdue ink mist. Later, the group inspected the installation.

EVANSVILLE (R) . . . Details of the construction and operation of a 3-ton, free piston refrigerator which uses natural gas as a fuel were given to members in May by Robert King, Battelle Memorial Institute. The description included estimated costs of building and operating such a system for domestic air conditioning, and also, the design of the lubrication and suspension systems.

ILLINOIS (H) . . . Chicago members, whose city is one of the remaining few where city water is used for condensing purposes and then disposed of, heard Walter M. Hassenplug cover briefly three basic systems for condensing purposes. Speaking at the April meeting, the Director of Engineering, Acme Industries, mentioned cooling towers, evaporative condensers and air cooling condensers. The speaker

(*Continued on page* 106)

	June		June		June
Miami Valley (H)	—	Ontario (H)	—	Southern Alberta (H)	—
Michigan (H)	—	Ontario (R)	5	Southern California (H)	—
Mid South (R)	—	Oregon (H)	—	Southern Connecticut (R)	—
Milwaukee (R)	2	Oregon (R)	—	Southern Piedmont (H)	—
Minnesota (H)	—	Ottawa Valley (H)	—	South Florida (R)	9
Mississippi (H)	—	Pacific-Northwest (R)	—	South Texas (H)	19
Mobile (R)	—	Panama Canal Zone (R)	17	Southwest Texas (H)	16
Montreal (H)	—	Philadelphia (H)	11	St. Louis (H)	—
Montreal (R)	—	Philadelphia (R)	—	St. Louis (R)	—
National Capital (R)	—	Pittsburgh (H)	—	Switzerland (H)	—
Nebraska (H)	9	Pittsburgh (R)	—	Toledo (H)	—
New Mexico (H)	16	Puget Sound (H)	—	Twin City (R)	5
New Orleans (R)	—	La Ville de Quebec (H)	—	Utah (H)	—
New York (H)	—	Richmond (R)	—	Virginia (H)	—
New York (R)	6	Rochester (R)	—	Washington, D. C. (H)	—
Northeastern Oklahoma (H)	—	Rocky Mountain (H)	—	Western Canada (R)	22
Northern Alberta (H)	—	Sacramento Valley (H)	3	Western Massachusetts (H)	—
Northern New Jersey (R)	—	San Antonio (R)	9	Western Michigan (H)	—
North Ohio (H)	—	San Francisco (R)	—	Western Michigan (R)	—
Northern Piedmont (H)	—	San Joaquin (R)	16	Western New York (H)	—
North Jersey (H)	9	Savannah (H)	—	West Texas (H)	—
North Texas (H)	—	Shreveport (H)	—	Wichita (R)	—
Oklahoma (H)	—	South Carolina (H)	—	Wisconsin (H)	—

Candidates for ASHRAE Membership

Following is a list of 146 candidates for membership or advancement in membership grade. Members are requested to assume their full share of responsibility in the acceptance of these candidates for membership by advising the Executive Secretary on or before June 30, 1959 of any whose eligibility for membership is questioned. Unless such objection is made these candidates will be voted upon by the Board of Directors.

Note: * Advance † Reinstatement

REGION I

Massachusetts

POCHASK, J. S., Asst. Sales Mgr., Market Forge Co., Everett.

New Jersey

TADDEI, O. W., Prod. & Dvlpt. Engr., Amerio Refrigerating Equipment Co., Inc., Cliffside Park.

New York

DARLING, R. B.,* Pres., Robert B. Darling Co., Inc., New York.
DIEHL, G. H., Chief Appl. Engr., Delco Appliance Div., General Motors Corp., Rochester.
DODGE, H. A.,† Cons. Engr., New York.
FEINER, BURTON, Sales Engr., Associated Thermal Products, New York.
GODFREY, J. W.,* Asst. Chief Engr., Harrison Radiator Div., General Motors Corp., Lockport.
HOFMANN, E. H., Mech. Engr., Voorhees Walker Smith Smith & Haines, Long Island City.
HOROWITZ, FRED, Designer II, The Port of New York Authority, New York.
MALITZ, STANLEY, Designer, Voorhees Walker Smith Smith & Haines, Long Island City.
MARK, RICHARD, Forest Prod. Engr., Balsa Ecuador Lumber Corp., New York.
MEHALICK, J. R.,* Sr. Engr., Carrier Corporation, Syracuse.
MEISENZAHL, T. W., Engr., Cherne & Dickson, Rochester.
NEUMAN, R. L., Sales Engr., Johnson Service Co., East Syracuse.
RANNOW, R. L., Sales Engr., York Corp., New York.
SCHMITT, C. A., Designer, Fred S. Dubin Assocs., New York.
THULL, G. A., Sales Engr., Johnson Service Co., Albany.
VOGEL, C. T., Sales Engr., Johnson Service Co., New York.

Rhode Island

BRINZA, A. B., Repr., National Plumbing & Heating Supply Corp., Providence.
JAMES, J. Z., Power Salesman, The Narragansett Electric Co., Providence.

REGION II

Canada

ALCOCK, G. W., Sales Engr., Canadian General Electric Co., Ltd., Toronto, Ont.
ALGIE, W. L.,* Mgr., Crane Associates & Warden King Ltd., Winnipeg, Man.
CAWLEY, R. H., Jr. Engr., Siemens Engineering Ltd., Edmonton, Alta.
GAUCHER, JULES, Tech. Adviser to Sales, Volcano, Ltd., Montreal, Que.
MCBRIDE, J. M., Pres. & Managing Dir., Industrial Power Installations, Ltd., Edmonton, Alta.
MCCAFFERY, R. A., Proj. Engr., Surveyer, Nenniger & Chenevert, Montreal, Que.
VODSEDALEK, R. A., Pres., Ottawa Mechanical Services Ltd., Ottawa, Ont.

REGION III

Pennsylvania

BUB, R. A., Engrg. Supvsr., Fluid Purification Section, Mine Safety Appliances Co., Pittsburgh.
BUCK, R. J., Dvlpt. Engr., York Div., Borg-Warner Corp., York.
BUNTEN, P. H., Vice-Pres., Sales, All American Metal Products Co., Inc., Philadelphia.
DOLHEIMER, R. E., Assoc. Appl. Engr., York Div., Borg-Warner Corp., York.
FLEURY, P. A., IV, Dist. Mgr., Chrysler Corp., Airtemp Div., Upper Darby.
HARRIS, G. L.,* A-C Engr., The M. & T. Co., Philadelphia.
NUTTING, R. E., Tech. Repr., Chrysler Corp., Airtemp Div., Upper Darby.
OVERSTREET, J. D., Jr., Sales Engr., American Radiator & Standard Sanitary Corp., Industrial Div., Philadelphia.
WOLFERT, J. E., Tech. Repr., Chrysler Corp., Airtemp Div., Upper Darby.
WOOD, SETH, Partner, Thermal Specialties Co., Harrisburg.

Virginia

DORSEY, J. B., Sales Engr., Johns-Manville Sales Corp., Richmond.
PORTER, J. B., Br. Mgr., Powers Regulator Co., Richmond.

REGION IV

Florida

DECKERT, C. P., Charge of Hgt. & A-C Dept., Hawthorne Roofing Co., W. Palm Beach.
LARSEN, L. F., Mgr., Hewett's Heating & Air Conditioning, Inc., St. Petersburg.
NICHOLS, R. L.,* Sales Engr., Worthington Corp., Miami.

North Carolina

SPEER, J. F., Sales Engr., American Radiator & Standard Sanitary Corp., Industrial Div., Charlotte.

South Carolina

HANNA, R. G., JR., Chief Engr., Reamer Industries, Inc., Columbia.

REGION V

Indiana

GAITHER, F. J., Asst. Sales Mgr., Jenn-Air Products Co., Inc., Indianapolis.

Ohio

KIRLIK, GEORGE, Proj. Engr., Standard Oil of Ohio, Cleveland.
PRINCE, F. J., Mgr. Sales & Engrg. Service, Perfection Industries, Cleveland.
PULLEY, T. R., Design Engr., Ranco, Inc., Columbus.
WHITE, J. A., Partner, Marlin White & Sons, Fremont.

REGION VI

Illinois

WILSEK, R. G., Sales Tech., Johnson Service Co., Chicago.

Iowa

MCKAY, J. K., Eastern Dist. Mgr., R. S. Stover Co., Marshalltown.

Michigan

BURTON, H. V., Sales Engr., John J. Nesbitt, Inc., Detroit.
CRUSE, E. A., Sales Engr., Air Systems, Inc., Detroit.
DEKORTE, C. W., Sales Prom. Engr., Michigan Consolidated Gas Co., Grand Rapids.
KOWALCZYK, H. J., Mech. Engr., Ford Motor Co., Plant Engrg. Dept., Dearborn.
OWEN, V. H., Repr., Union Steel Products Co., Albion.
SCHREIBER, W. A., Asst. Prof., A-C., Western Michigan University, Kalamazoo.
SENKOWSKI, R. M., Mech. Engr., Lee Paper Co., Vicksburg.
TIBBITS, C. B., Mech. Engr., Ford Div., Ford Motor Co., Dearborn.
WOLF, R. E., Gen. Mgr. & Engr., Wolf Sheet Metal & Specialty Co., Kalamazoo.

Minnesota

FEILZER, J. E.,† Sr. Mech. Engr., City of Minneapolis, Minneapolis.
FREDERICK, W. J., Sales Engr., Northern States Power Co., Minneapolis.

Wisconsin

LARSON, K. G., Vice-Pres., Gustave A.

Larson Co., Milwaukee.
ROEDEL, R. D., Sales Engr., Steam Plant Equipment Corp., Milwaukee.

REGION VII

Alabama

ESKEW, T. M., Partner, H. L. Eskew & Sons, Birmingham.
SIBLEY, B. P., Appl. Engr., Cutler-Hammer, Inc., Birmingham.

Missouri

KOTTWITZ, W. H., Engr., Belt & Given, St. Louis.

Tennessee

ALLEN, W. H.,† Owner, Harwell Allen & Assocs., Nashville.
ANGEL, R. P., Service Engr., The Kroger Co., Nashville.
BERKENSTOCK, H. R., Proj. Engr., Heil-Quaker Corp., Nashville.
BIBB, R. L., JR.,† Vice-Pres., Nashville Machine & Supply Co., Nashville.
DILLINGHAM, D. D., Constr. Supt., M. T. Gossett Co., Inc., Nashville.
GARDNER, R. E., Prop., R. E. Gardner Co., Nashville.
KEMP, B. G., Br. Mgr., American-Standard, American Blower Division, Nashville.
KENNEDY, EDDIE, Vice-Pres., Ed's Supply Co., Nashville.
LEE, W. L., Sales Engr., Lee Refrigeration Co., Nashville.
LONGHURST, L. A., JR., Engr., M. T. Gossett Co., Nashville.
MATHIS, J. B., Chief Proj. Engr., Heil-Quaker Corp., Nashville.
MAYER, R. E., Proj. Engr., Heil-Quaker Corp., Nashville.
MCCAULEY, L. C., Sales Mgr., J. A. Harwell & Assocs., Nashville.
MOATS, E. W., Vice-Pres., M. T. Gossett Co., Inc., Nashville.
NANCE, W. E., Pres., Ed's Supply Co., Nashville.
NICHOLS, D. C., Assoc., I. C. Thomasson & Assocs., Nashville.
NORTH, J. E., Proj. Engr., Heil-Quaker Corp., Nashville.
ORR, D. C., Supt., Nashville Machine & Supply Co., Nashville.
PEARL, J. H., Repr., J. B. Thomas Co., Nashville.
RADLEY, ERNEST, JR., Proj. Engr., Heil-Quaker Corp., Nashville.
REID, A. W., Designer, I. C. Thomasson & Assocs., Nashville.
ROWLETTE, W. M., Engr., Andrews Distributing Co., Inc., Nashville.
THOMASSON, I. C., Owner, I. C. Thomasson & Assocs., Nashville.
TOLLEFSON, C. N., Engr., Climate Control Co., Nashville.
TOWLE, P. S., Sales Engr., Dunham-Bush Inc., Fountain City.
TRAVIS, E. B., Pres., Air Conditioning Sales & Service, Inc., Nashville.
VICK, H. W., JR., Sales Engr., The Trane Co., Nashville.
WILHOITE, MURRAY, Chief Proj. Engr., Temco Inc., Nashville.
WOOD, R. H., Repr., Minneapolis-Honeywell Regulator Co., Nashville.

REGION VIII

Arkansas

RULE, J. H., JR.,† Design Engr., Erhart, Eichenbaum, Rauch & Blass, Little Rock.

Oklahoma

DILBECK, J. D., Sales Engr., Norman Plumbing Supply Co., Oklahoma City.
NELSON, R. S., Sales Engr., American Radiator & Standard Sanitary Corp., Industrial Div., Oklahoma City.

Texas

GAMBRELL, J. N., JR., Sales Engr., The Trane Co., Houston.
HERRIDGE, K. H., Owner & Mgr., K. H. Herridge Maintenance Co., Houston.
LOCKERBY, W. L., Owner, Lockerby Engineering Co., Houston.
SEXTON, G. E., Gen. Mgr., Climate Supply Co., Inc., Dallas.
WILHELM, J. K., Mech. Proj. Engr., Lockwood Andrews & Newman, Houston.

REGION IX

Colorado

BEAUCHAMP, R. C., Dist. Mgr., Pacific Pumping Co., Denver.

Kansas

REA, J. D., Owner, J. D. Rea Heating & Air Conditioning, Wichita.

New Mexico

RAY, K. L., Mech. Designer, J. L. Breese & Assocs., Santa Fe.

Utah

WILLIAMS, R. W., JR., Partner, Rex Williams & Sons, Salt Lake City.

REGION X

Arizona

GOODRICH, R. C., Br. Mgr., Ware-McClelland Supply Co., Tucson.
HERMANN, W. J., Engr., John Paul Jones, Tucson.

California

BRITT, L. O., Vice-Pres., Gen. Mgr., Dudley Deane & Assoc., San Francisco.
CRIVELLO, S. V., Design Engr., Frank L. Hope & Assocs., San Diego.
GAMBLE, A. E., JR., Tech., Johnson Control Co., Los Angeles.
HECKMAN, A. R., Mech. Engr., Daniel, Mann, Johnson & Mendenhall, Los Angeles.
HUNTINGTON, R. B., Vice-Pres., University Mechanical & Engineering Contractors, San Diego.
KELLEY, G. F.,* Owner, Kelley Air Conditioning Engineering, Modesto.
MAROKO, M. C., Cons. Mech. Engr., Samuel L. Kaye, Los Angeles.
REICH, JACK, Mech. Designer, J. S. Hamel Engrg., Inc., Burbank.
RESTAINO, PAT, Sales Engr., Minneapolis-Honeywell Regulator Co., Sacramento.
RICKARD, R. C., Proj. Engr., Monrovia Aviation Corp., Monrovia.
SAUER, E. I.,* Mech. Engr., N.O.T.S., United States Navy, China Lake.
SCHULER, G. F., III, Co-Partner, George F. Schuler & Son, Stockton.
STAUCH, W. A., Mech. Engr., Hughes Aircraft Co., Fullerton.
THOMPSON, EZRA, Br. Mgr., Powers Regulator Co., Sacramento.

Oregon

CREEGAN, J. V., Mgr., Low Temp Insulation Dept., Fiberglas Engineering & Supply, Portland.
MCCORMICK, G. L., Estimator, H. W. McKenzie Co., Portland.
URBEN, C. L., Mech. Engr., J. Donald Kroeker & Assocs., Portland.

Washington

BARKER, T. R., Sales Engr., Minneapolis-Honeywell Regulator Co., Spokane.
DAPPER, M. J.,† Mgr., Hgt. Div., Fuel Oil Service Co., Tacoma.
MASSART, G. F., JR., Estimator, Grant County Plumbing & Heating Co., Moses Lake.
MCCULLOCH, E. W., Appl. Engr., Minneapolis-Honeywell Regulator Co., Spokane.

U. S. POSSESSIONS

Canal Zone

PAYNE, I. M., Inspector, Gen. Constr., U. S. Army, Caribbean, Office of the Engineer, Fort Clayton.
PRETZ, C. E., Lead Foreman (Refrigeration), Panama Canal Co., Balboa Hgts.
STEWARD, J. W., Owner, Steward Electric & Refrigeration, Curundu.

FOREIGN

Australia

BLUNT, M. G., Managing Dir., M. G. Blunt, Pty., Ltd., Queensland.

England

HAZELL, K. J., JR., Design Engr., Carrier-Ross Engineering Co., Ltd., London, W. I.

India

OZA, B. C., Executive Asst., Air Conditioning Corp. (P) Ltd., West Bengal.

Italy

GALLINO, CARLO, Designer, Star Aeromeccanica Corp., Turin.

Jamaica

BIERMANN, G. T., Gen. Mgr., A-C. & Refr. Div., J. S. Webster & Sons, Ltd., Kingston.

Singapore

CUTTS, W. H., Installation & Maint. Engr., Sime Darby (Singapore).

South Africa

SAMUELS, SYDNEY,* Cons. Engr., Royton Electrical Engineering Co., Johannesburg.

South America

SCHEFFEL, W. C., Engr. A-C. Dept.,

International Engineering Consultants NV, Surinam.

South Rhodesia

ELLISON, T. M., Managing Dir., Ellisons Electrical Engineers (Pvt.) Ltd., Salisbury.

Switzerland

BERCHTOLD, PETER, Partner, Berchtold & Co., Thalwil.

Venezuela

FERRER, JOSÉ, Design Engr., S.A.V.-E.R. Guinand, Caracas.

STUDENT CANDIDATES

AVANZADO, M. B., University of Wisconsin, Madison.

BULLETINS

(Continued from page 79)

process two narrow rolls, and facilities for each record removal.
Consolidated Electrodynamics Corporation, 360 Sierra Madre Villa, Pasadena, Calif.

Timing Belt Drive. Steel cable cords imbedded in neoprene fully molded with nylon base fabric go into the construction of timing belt drives, as described and illustrated in this Bulletin 20B8760. The drive is considered suitable for a wide range of load applications, from sub-fractional to 600 hp with a torque load range from thousands of ft-lb to light load in.-oz.
Allis-Chalmers Manufacturing Company, Milwaukee 1, Wisc.

Corrugated Expansion Joints. As a guide to expansion joint selection, this Bulletin includes self-equalizing, duo-equalizing, non-equalizing, flexible connectors, high pressure toroidal, universal, modified universal, hinged and gimbal, pressure balanced and pressure balanced universal expansion joints.
Zallea Bros., 815 Locust St., Wilmington 99, Del.

Summer Systems. Air conditioning systems for summer use, in this air cooled line, are described in Flyer No. 165.
Henry Furnace Company, Medina, Ohio.

Tube Mill Products. Illustrations of a tube manufacturing plant and steps in the manufacture of the product are part of this Bulletin which concludes with a 2-page applications listing and chart.
Scovill Manufacturing Company, Mill Products Div, 99 Mill St., Waterbury 20, Conn.

Manual Brazing. Hand torches for silver alloy brazing are illustrated with text on various applications in 4-page Bulletin 81.
Handy & Harman, 82 Fulton St., New York 38, N. Y.

Vibration Measurement and Control. Eight-page Bulletin K4E gives engineering specifications and performance data for 27 types of products for the control and measurement of machinery vibration, shock and noise. A detailed discussion of the relative merits of steel springs and organic materials as isolation media is included.
Korfund Company, Inc., 48-530 32nd Place, Long Island City, 1, N. Y.

Packaged Liquid Chillers. Specifications and explanations of liquid chillers in capacities of 20 to 180 ton are illustrated in this Bulletin, 802-A.
Frick Company, Waynesboro, Pa.

Circuit Breaker. Folder CIRB-24 gives the fundamental design and actual size illustration of a circuit breaker which eliminates the use of several springs and adjusting pins which are prone to disorder. With ratings at intervals from 5 to 25 amp,

(Continued on page 86)

Some specific projects

BURGESS H. JENNINGS
Director of Research
ASHRAE Research Laboratory

Research Laboratory Meetings – During the spring, the Laboratory has been host to a large number of Committees. At the time of writing this copy, recent meetings included a meeting of the Research and Technical Executive Committee, April 10; the Combustion Conference Guiding Committee, April 16; the RAC on Hot Water and Steam Heating, April 22; the RAC on Odors, April 29; and the RAC on Insulation, April 30 and May 1.

The Conference on Combustion in the Heating Industry was held in Cleveland, May 21. At the Conference, which was well-attended, eleven panelists discussed various phases of combustion, ranging from the atomization-mixing properties of fuels and their proper combustion to catalytic combustion and fuel cells. A detailed summary of the Conference is being prepared for a subsequent issue of the JOURNAL.

The Research Advisory Committee meetings brought to light a number of interesting topics. At the Hot Water and Steam Heating meeting discussion relating to the research project now in progress on Steam Flow indicated that a word of caution should be sounded on the use of steam-flow data. Flow charts are based on the flow of dry saturated steam throughout the length of line. In practice, condensation occurs in the pipe, and the actual flow at the inlet end is equal to the steam delivered plus the condensate. This means that for a given delivery of dry steam at the end of the line, the pressure loss will be greater than indicated by the chart. For larger pipe sizes and reasonable pipe lengths, the error will not be serious, but for long lines of small pipe, the error can be appreciable.

The need for better data on the flow of high-temperature water was discussed at some length. It was pointed out that in a low-temperature hot-water system designed for a 15 or 20 degree drop, the heat transfer rate will change but little with relatively large changes in flow rates. In contrast to this, the heat transfer rate in a high-temperature system varies more directly with the flow rate, and thus, the flow rate is much more critical. The RAC is making a careful study of the flow characteristics of high-temperature water.

Another interesting problem suggested at the meeting was concerned with the selection of safety valves for the low-pressure side of any water heater which receives its heat from a high-temperature water system. Valves as presently selected would not provide adequate protection in case a tube ruptured within the heat exchanger.

The meeting of the RAC on Odors made a series of recommendations of research projects for that Committee, and plans are being rapidly started to carry out a comprehensive program in this field.

Sound Research Progress – One phase of the sound-research program under the direction of William Kerka, and being carried out under a grant from the Bureau of Ships of the U.S. Navy, has been completed. As this should be of some interest, a few comments concerning it will be made here.

In the past, the determination of sound levels within ducts has taken little note of the noise generated by the system components due to air flow. While a fan may be the primary source of noise, some of this sound will be attenuated by the elbows, filters, and coils of the system. As the high-velocity air passes through these components (and for that matter, over any projection into the air stream), each in turn serves as an individual noise generator. As a result, a point in the duct system may be reached where the noise-generation level of the components begins to exceed the attenuated sound level from the fan. To date, most published data on the acoustic performance of elbows, commercial sound traps and terminal devices are based on their sound-attenuation performance only, and were probably determined with speaker sound sources. With high-velocity systems becoming more prevalent, however, information on the noise-generation of the components must be known to predict more accurately the expected levels within the duct, and ultimately, within the occupied space.

A recent study at the ASHRAE Research Laboratory in Cleveland has investigated both the noise-generation and attenuation characteristics of elbows with turning vanes. The noise-generation studies were made at air velocities ranging from 2000 to 5000 fpm. A correlation of the sound attenuation and generation in elbows versus size, aspect ratio, and angle has also been made. Because an elbow (as normally mounted) is both acoustically and physically coupled to adjacent straight duct, studies of the sound attenuation in unlined and uninsulated ducting of different sizes and aspect ratio were also conducted. A final report on this work will soon be released.

it is designed primarily for remote, overcurrent protection of 110 volt ac motors through ½ hp.
Metals & Controls Corporation, Attleboro, Mass.

Limited Space Motor. Rated from 1 to 5 hp in dripproof and totally enclosed constructions for limited space applications, this Thinline motor is the subject of eight page Bulletin GEA-6927. The illustrated text includes description of product features, chart of interchangeable flange dimensions and data on both dripproof and enclosed models.
General Electric Company, Schenectady 5, N. Y.

Air Purification Equipment. Data on activated carbon air purification equipment is given in Bulletin 108A, 12 pages.
Connor Engineering Corporation, Danbury, Conn.

Noise Measurement. Non-mathematical in approach, this handbook of noise measurement defines sound and sound pressure levels before explaining sound-survey meters, portable battery-operated instruments, and their usage.
General Radio Company, West Concord, Mass.

Space Filter. Performance, construction and installation details of the Dustfoe space filter are covered in 8-page Bulletin 1505-6.
Mine Safety Appliances Company, 201 N. Braddock Ave., Pittsburgh 8, Pa.

Portable Moisture Monitor. Two types of instruments for monitoring or control of moisture in gases or gaseous mixtures are the subject of Bulletin 1834C. Applications, features, operating characteristics, specifications and other details are described in 8 pages.
Consolidated Electrodynamics Corporation, 360 N. Sierra Madre Villa, Pasadena, Calif.

Clutch-Pulley. Descriptive of a clutch-pulley package for direct installation on standard electric motors from 1 to 25 hp, this 10-page Electro-Sheave Brochure is numbered WEB P-52.
Warner Electric Brake and Clutch Company, Beloit, Wisc.

Gear and Fluid Drives. Functions of the various types of drives in a line of Gearmotors, Motogears and Fluid Drives, and detailed selection data, dimensions, overhung load ratings and mountings are given in this 48 page Catalog 2747. A total of 45 new units has been added to the line, which now has link belt motogears and gearmotors in quadruple as well as double and triple reduction units, permitting ratios from 6.2:1 through 985:1. Larger sizes have also been added and capacities now range up to 100 hp with output speeds from 280 rpm down to 1.8 rpm.
Link-Belt Company, Prudential Plaza, Chicago 1, Ill.

Coil-in-Plate Heater. Made of a wide variety of metals, this Thermo-Panel Coil, as cited in Price Data Bulletin 259, is offered as an improvement over pipe coils. Four pages.
Dean Products, Thermo-Panel Coil Div, 616 Franklin Ave., Brooklyn 38, N. Y.

Small Valves. Information pertaining to working pressures, specifications, construction features and suggested applications is compiled in this four page brochure, KS-1-N.
Kerotest Manufacturing Company, 2525 Liberty Ave., Pittsburgh 22, Pa.

Central Air Conditioners. Bulletin 9227 describes a line of low pressure and high pressure central station air conditioners. Design features, construction details, cooling and heating coils and accessories for the Types A, S, AB and C units that make up the line are given. Tables are included to indicate fan capacities, coil air friction, selection factors, water coil data, coil capacities and physical data.
American Radiator and Standard Sanitary Corporation, Industrial Div, Detroit 32, Mich.

Centrifugal and Rotary Pumps. Open-impeller centrifugal pumps, both sealed and sealless types, and mechanical positive-displacement rotary pumps are the subject of this 72-page catalog. The pumps vary in size from 1/20 to 5 hp; delivering from 1 pt to 148 gpm, and operating at pressures of from zero up to 54 psi.
Pioneer Pumps, Detroit Harvester Company, 21800 Greenfield Rd., Oak Park 37, Mich.

Resistance Temperature Elements. For measurement of temperatures to 1200 F under pressures to 4500 psig, these resistance elements are described in 4-page Product Specification E51-6. Selection charts are included along with graphs showing limitations of protecting wells and leadwires.
Bailey Meter Company, 1050 Ivanhoe Rd., Cleveland 10, Ohio.

Custom Molded Plastics. Explanation of how to buy plastics custom moldings precedes a fourteen-question check list designed to assist manufacturers who want to investigate the use of custom-molded plastics parts and components.
Monsanto Chemical Company, Springfield, Mass.

Research Chambers. Altitude walk-in rooms, temperature-humidity walk-in rooms, chillers and equipment for testing under conditions of sand and dust are covered in 12-page Catalog 59.
American Research Corporation, Farmington, Conn.

Steel Cooling Towers. In 2 to 120 ton capacity, these horizonal induced draft steel cooling towers, as described in Bulletin 53-902, feature a self-cleaning sloping floor basin which will clean itself of debris.
J. F. Pritchard & Company of California, 4625 Roanoke Parkway, Kansas City 12, Mo.

Others are saying

(Continued from 77)

In the UNESCO Secretariat in Paris, the principal sunshading device is a ribbon of glass combined with horizontal louvered concrete overhangs. Solar studies, summarized here, showed the designers how far the glass sunshades, which have a constant depth, should project from the overhang to get equivalent shading on the various orientations when the sun's radiation is at its peak. *Architectural Record, March 1959, p 226.*

that . . . . . . prolonged submergence of today's submarine has made habitability an important factor in crew members' well-being. Some of the problems have been solved, but investigation into the effects of condensation droplets and positive and negative ions has revealed that these may have an effect on respiration, and hence morale and efficiency, of personnel. The data presented here goes beyond submarines. It would apply to any occupied closed structure housing ion generating materials. *Heating, Piping and Air Conditioning, May 1959, p 101.*

PARTS AND PRODUCTS

GAS CONVERSION BURNERS

Power (forced draft) inshot gas conversion burners in the Contractor series are in two basic models: 75,000 to 225,000 Btu and 150,000 to 400,000 Btu.

Burners in 5, 8, 10 and 15 in. usable blast tube lengths have tube diameters of 4 in. and are provided with either adjustable flange or stand mountings.

An adjustable orifice permits exact setting of Btu input by turning the needle type orifice assembly with a special tool provided. Separate adjustment of primary air and secondary air allows the installer to adjust the flame exactly to the combustion chamber.

Barber Manufacturing Company, Cleveland, Ohio.

AIR OVER FAN MOTORS

In ratings from 1 to 125 hp, this line of motors is designed for quiet operation in ventilating systems, exhaust systems, cooling towers and all air moving installations where a motor drives a propeller or axial flow fan. The motor design makes use of air circulated in these installations to cool the motor so that maximum hp may be produced from a minimum motor size.

Motors are with foot or flange mountings, in totally enclosed and explosion proof enclosures, and are suitable for vertical or horizontal mounting.

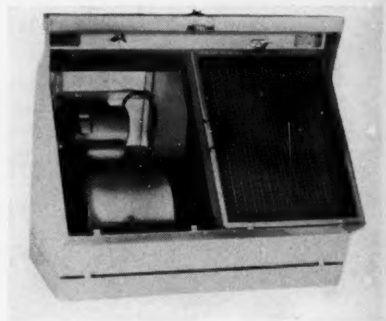
Louis Allis Company, Dept. P, Milwaukee 1, Wisc.

RANGE VENTING SYSTEM

Designed for restaurants and other mass feeding establishments, this range venting system requires no roof fan installation. It is a self-contained unit complete with blower and motor.

Identified as the Filtaire, the unit minimizes smoke, heat and odors by reduced blower pressure directly over the range.

Installation is effected by bolting the adapter case into the exhaust



hood, and hinging and locking the unit into the adapter. When unlatched, the entire unit swings down and forward for ease of servicing.

Single, double and triple filter units are in capacities of 1000, 2000 and 3000 cfm.

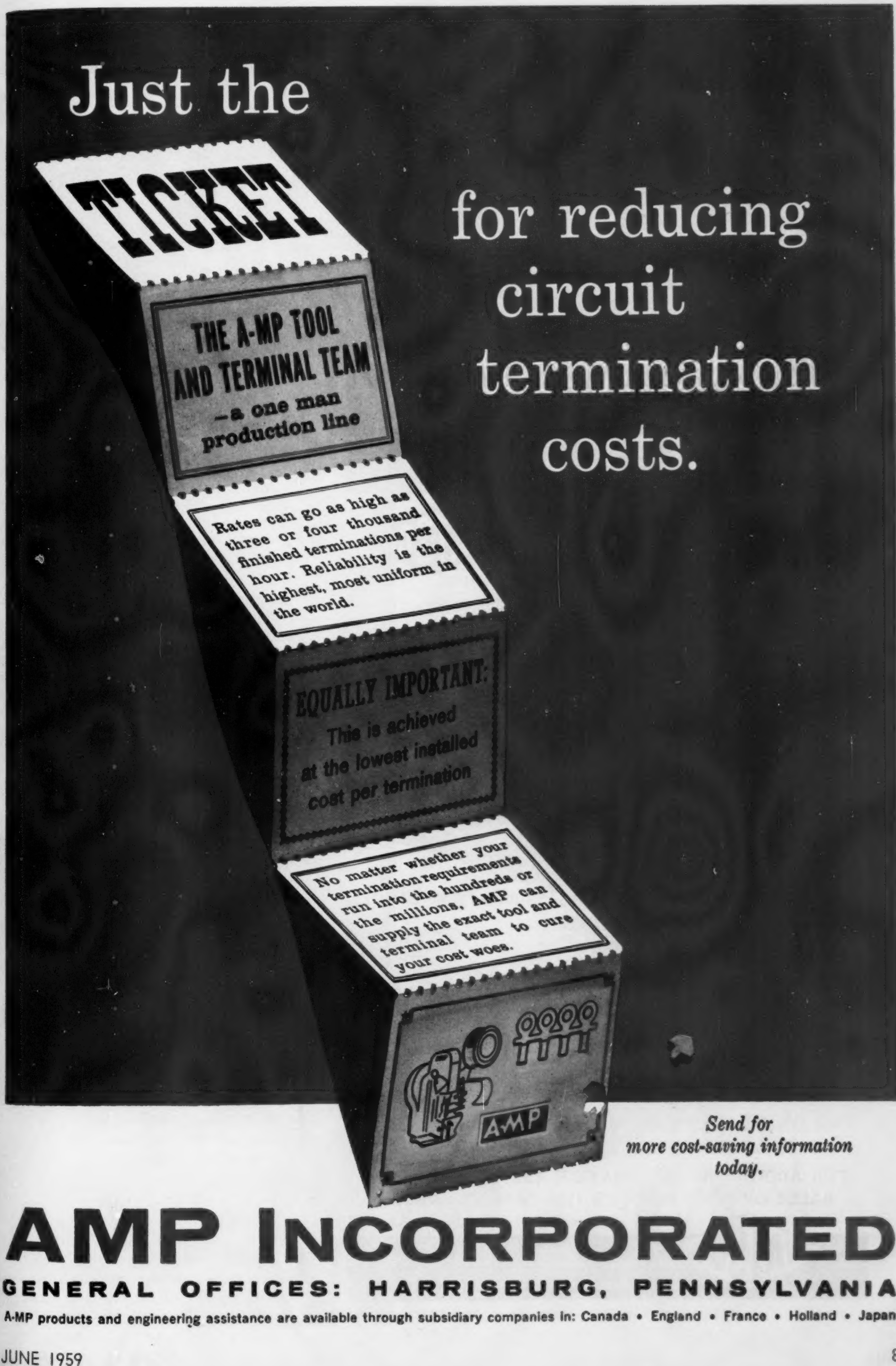
Morrison Products, Inc., 16816 Waterloo Rd., Cleveland 10, Ohio.

LIQUID ALGAECIDE

To protect cooling towers and evaporative condensers from the insulating effect of slimy growths on heat transfer surfaces, this Biocide RP will resist leaching out by water for as long as two months after first being introduced into the system.

A liquid inhibitor, it will control many types of microorganisms, including algae, bacteria and fungi, and will eliminate the clogging of holes in distribution pans and pump intake screens.

One oz of the biocide per 50 gal of water is cited as effective enough to kill most types of microorganisms. It is slug fed or dumped into the system at regular intervals ranging from once a week to every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TICKET
for reducing circuit termination costs.
THE A-MP TOOL AND TERMINAL TEAM
—a one man production line
Rates can go as high as three or four thousand finished terminations per hour. Reliability is the highest, most uniform in the world.
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This is achieved at the lowest installed cost per termination
No matter whether your termination requirements run into the hundreds or the millions, AMP can supply the exact tool and terminal team to cure your cost woes.
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GENERAL OFFICES: HARRISBURG, PENNSYLVANIA
A-MP products and engineering assistance are available through subsidiary companies in: Canada • England • France • Holland • Japan

two months. Rate of feed depends on how fast growth occurs, which in turn, is affected by variables such as type of microorganism, contaminants in the water, pH and temperature.
Calgon Company, Box 1346, Pittsburgh 30, Pa.

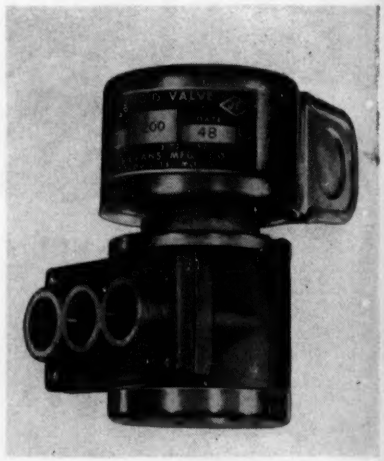
HOUSE TRAILER DUCTING-INSULATION

Rigid urethane foam is both duct and insulation in an air duct devised for mobile homes. Heat loss from the source to the outlets is cited as less than 6%, although the full length of the 33 ft duct is exposed to outside weather. The ¾ in. thick duct may also be used as an air conditioning system with cold air.

Stafoam urethane, poured in place, is used in the aluminum foil-lined ducts which carry warm air from a central heating system to floor registers, the full length of the home. Conventional duct material is insulated sheet metal.
Dayton Rubber Company, Plastics Div, Dayton, Ohio.

3-WAY SOLENOID

Water control for fan-coil units, using forced circulation chilled or hot water for individual room air conditioning, is effected by this 3-way solenoid valve which diverts water either through the heat exchange coil or to



by-pass the coil when temperature requirements are met. Capacity range is from ½ to 10 gpm with a maximum operating pressure of 25 psi (200 psi static).

The Type 701 solenoid utilizes pilot operation and diaphragm amplification, and is cited as providing maximum flow at low pressure and eliminating fluid hammer and electrical hum.
Jackes-Evans Manufacturing Company, Controls Div, 4427 Geraldine Ave., St. Louis 15, Mo.

MOISTURE INDICATOR LAMP

Relative humidity content of indoor atmospheres may be determined by this compact device which plugs into any electrical socket and indicates the per cent of moisture in the air.

An electronic sensing element in the Moisture-Lite-Indicator gives an instantaneous response to changes of moisture in the air. Rh is indicated by a tiny electric lamp which first begins to glow slightly when rh reaches about 30%. A soft glow indicates a balance of moisture (35-50%) and as rh goes higher, the lamp becomes brighter.

When exposed to outside weather through open doors or windows in the summertime, the lamp will indicate rain or storm; placed in the basement when the furnace is idle, it will indicate dampness.
Newmack Corporation, 147 Eady Court, Elyria, Ohio.

EVAPORATOR COILS

Installed in connection with the Moncrief 3 or 5 hp air cooled condenser-compressor unit, these counterflow evaporators develop cooling capacities of 36,000 or 56,500 Btu/hr, respectively. (Ratings are based on

standard test conditions of 80 F DB and 67 F WB, 95 F outdoor air temperature.)

The evaporator is designed and constructed to provide maximum drainage of condensate moisture. A flat coil, it is mounted horizontally and on a slight slant within a rigid steel frame. A series of zinc coated drainage trays, which is suspended beneath the coil, collects the condensate water and carries it to a built-in drain pan.

As an accessory, an insulated enameled 16-gauge steel cabinet provides slide-in installation and ready accessibility for the evaporator, plus rigid support for the furnace which is installed on top of it.
Henry Furnace Company, Medina, Ohio.

GAS WALL HEATER

Three-way installation is the feature of this direct-vent gas wall heater in the winter and summer air conditioning line. The Custom Counterflow may be installed fully recessed, partially recessed or hung on the wall.

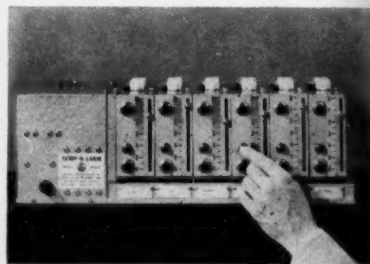
Vents are flush with the outside wall in any of the three positions.

Utilizing the counterflow principle, the unit delivers heat at the floor. Only outside air is used for combustion and products of combustion are exhausted to outside through the sealed porcelain combustion chamber. Capacity is 30,000 Btu/hr input.
Heil-Quaker Corporation, 647 Thompson Lane, Nashville, Tenn.

DEFROST ALARM

Triggering an alarm when temperatures in frozen food cabinets reach thaw level, this equipment comprises a thermostat pre-set to actuate at any desired temperature. When the temperature in a freezer or chill box reaches the pre-set temperature, it sends a signal to an annunciator which causes audible and/or visible alarms to sound either locally, at a central alarm station, or both.

An alarm time delay may be set from zero to six hr to minimize the possibility of alarms being set off



during expected and normal thaw temperatures as a result of loading or defrosting. Temp-O-Larm thermostats have a ± 5 F variance.

One annunciator drive unit offered will monitor from one to six chill or freeze cabinets and another will monitor seven to twelve.
Kidde Ultrasonic & Detection Alarms, Inc., Clifton, N. J.

INLINE DIFFUSERS

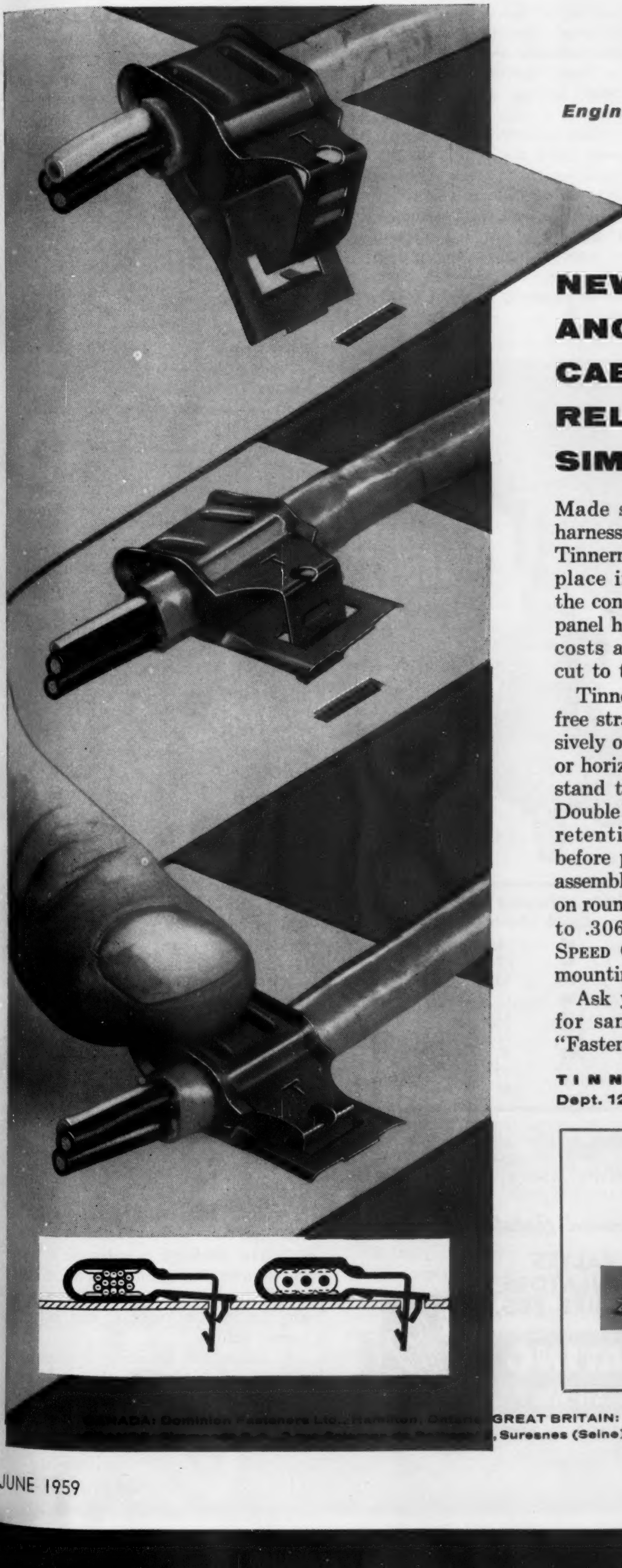
One-way blow (KM-1) and two-way blow (KM-2) in-line diffusers are in a number of widths and lengths, with spring and catch design providing easy removal or installation of large sections without tools. They are part of this manufacturer's K series of ceiling and wall diffusers.
Carnes Corporation, Verona, Wisc.

BLOWER-EVAPORATOR TRUCK UNIT

Closely-controlled defrosting and wide fin spacing are features of the Super 35 blower-evaporator developed for performance in medium temperature truck bodies up to 14 ft long.

Mounted on the front, rear or side wall, the 12 x 15½ x 51 in. blower is in a ribbon design to preserve load space and aisle space.

Automatic defrosting of the unit



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NEW SPEED CLIP® ANCHORS WIRES, CABLES, TUBING, RELIEVES STRAIN, SIMPLIFIES ASSEMBLY

Made specifically to attach cables, wires, harness, or tubing firmly to panels, this newest Tinnerman SPEED CLIP is readily snapped into place in only 3 simple steps. Prelatch it on the conductor or tubing, insert SPEED CLIP in panel hole, then push home to lock. Assembly costs are reduced because assembly time is cut to the minimum.

Tinnerman SPEED CLIPS also serve as trouble-free strain-relief clamps—they are used extensively on appliances for attaching 3-wire round or horizontal section rib cord, and easily withstand the 35-pound pull test requirements. Double latch permits pre-assembly and accurate retention of SPEED CLIPS to wire or harness before panel assembly for further savings in assembly time. Double-rib retainers grip tightly on round or rectangular cords from .175″ round to .306 x .515″ rectangle. Important, too, SPEED CLIPS can easily be removed from the mounting side.

Ask your Tinnerman sales representative for samples and prices. He's listed under "Fasteners" in most Yellow Pages. Or write to:

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Speed Nuts®

FASTEST THING IN FASTENINGS®

CANADA: Dominion Fasteners Ltd., Hamilton, Ontario. GREAT BRITAIN: Simmonds Aerocessories Ltd., Treforest, Wales. ..., Suresnes (Seine). GERMANY: Mecano-Bundy GmbH, Heidelberg.

is controlled by a de-icer thermostat and positive-action solenoid valves. The coil is defrosted with hot gas only when it needs defrosting and controls will not allow the refrigeration cycle to start until the coil is completely defrosted.
Kold-Hold Div, Tranter Manufacturing Company, 735 Hazel St., Lansing 9, Mich.

FLOW INDICATOR

For indicating the flow of water in gpm through a hot water heating or air conditioning installation, this improved hydronic device is described as having no moving parts to vibrate. Additional advantages of the Thermoflo Indicator are cited as low pressure drop, never more than two Monoflo fittings, and an easy-to-read scale.

Two soldered connections and a square head cock are all that are needed for installation. It is adjustable to 90 deg intervals, and in either a vertical or horizontal position. It is particularly suited for determining flow rates of pumps having a flat capacity curve.
Bell & Gossett Company, Morton Grove, Ill.

HIGH HEAD CIRCULATORS

Addition of a series of high head circulators to this line of heaters brings to six the number of sizes. With single or three phase non-overloading motors, they are specifically designed for heating and cooling, boosting water pressure and recirculation hot water applications.
Taco Heaters, Inc., Cranston, R. I.

ELECTRONIC CLEANER UNIT

This electronic air cleaner designed to fit nationally distributed furnaces and air conditioners is no larger than an ordinary disposable air filter, but is offered as being more efficient.

Identified as the DFL, it operates on the principle of electrostatic precipitation, with action similar to a magnet. As unclean air is circulated through the unit's aluminum ionizing frame section, the particles are given a positive electrical charge. Foreign material is then attached to a negatively charged disposable collecting pad. Clean air then passes through a second aluminum perforated frame and is recirculated through the building.

Capacities range from 640 to 1000 cfm per panel. Where necessary, the DFL cleaning unit may be installed in an adapter in the return air duct. A gauge indicates when the collecting pad should be changed.
Trion, Inc., McKees Rocks, Pa.

CHEMICAL CORROSION CONTROL

Inhibiting corrosion in closed recirculating water systems, with or without anti-freeze, this Corrosion Inhibitor CS is a non-chromate product which will protect steel, copper, brass, aluminum and solder.
Hagan Chemicals & Controls, Inc., P. O. Box 1346, Pittsburgh 30, Pa.

PACKAGE COMBINATION

Designed for roof-mounting, this packaged air-cooled air conditioner and gas-fired heater combination is in two sizes with nominal air conditioning capacities of 5 to 8 ton. It offers a complete year-round system for commercial and industrial buildings. The gas-fired heating system providing 160,000 Btu/hr output is contained completely within the unit.

The package consists of a cooling system, the gas-fired heating unit, air filters, blowers and controls. An air distribution plenum and diffuser comes with it.

Intake and discharge connections are provided through the end, permitting the unit to be placed over weight-bearing building members and

(Continued on page 97)

NEW PRODUCTS

(Continued from page 94)

running ductwork to the diffuser. Additional roof support is not usually required.

The 5-ton unit weighs 2000 lb, the 8-ton, 2200 lb; both measure 55 x 92 x 74.

Typhoon Air Conditioning Company, Div of Hupp Corporation, 505 Carroll St., Brooklyn 15, N. Y.

FLAKE ICE MACHINES

Mounted on the top or side of any bin, these high-capacity flake ice machines are 14 x 16 x 20 in. Both models, FL-100, which makes 96 lb of ice per day at the rate of 4 lb/hr with room temperature of 90 F, water temperature of 70 F; and FL-200, which produces 192 lb per day at the rate of 8 lb/hr; make hard, dry individual flakes of ice.

Ice production up to 5 lb/hr with the first model, and 10 lb/hr with the second may be expected with lower room and water temperatures.

Flake Ice Machines, Inc., Div of Helmco-Lacy, Chicago 31, Ill.

AIR HANDLING MODELS

Nine air conditioners added to this line include two 115-volt, 1 hp; a 1, 1½ and 2 hp in 208 or 230 volts; and a 1 hp heat pump.

Fedders Corporation, 58-01 Grand Ave., Maspeth 78, N. Y.

WATER COOLED CONDENSER

Featuring the shell and coil type of condenser-receiver assembly used on all water-cooled models of this manufacturer, Model BWRH33 is designed to conserve space for the installer. Over-all dimensions are 16¼ x 15¼ x 10½ in. It is supplied complete with dual pressure control and a water regulating valve. The unit incorporates a single cylinder compressor that is designed for use with Refrigerant-12 and motor characteristics of 115 volt, 60 cycle, single phase.

Bendix-Westinghouse Automotive Air Brake Company, 950 East Virginia St., Evansville 11, Ind.

TESTING CHAMBER

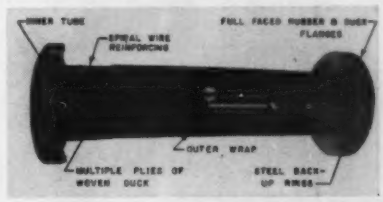
Utilization of liquid carbon dioxide in this environmental test chamber allows for rapid pulldown (from ambient to −100 F in less than 3 min.) and for the dissipation of huge heat loads within the chamber (at least 5000 Btu/hr at −100 F).

A multi-pane window as optional equipment provides a full view of the chamber interior, which is of welded stainless steel. Insulation is provided by 5-in. thick walls of high density glass fiber material, and the exterior is enameled galvanneal.

Associated Testing Laboratories, Inc., Manufacturing Div, Clinton Rd, Caldwell, N. J.

RUBBER PIPING CONNECTORS

All purpose rubber flexible piping connectors, for use in controlling water line noises, eliminating equipment vibration, reducing water hammer,



simplifying alignment problems and minimizing strain on piping, have an acoustical impedance far below that of steel pipe, and will attenuate up to 90% of vibration and noise.

Flex-Hose is constructed of a rubber liner, several layers of multiple-ply rubber and fabric, wire reinforcing, another layer of rubber and fabric, and an outer wrap. Special inner and outer rubber tubes provide resistance to chemical action and abrasion, and withstand temperatures up to 250 F, and handle pressures up to 250 psi.

Type RF with flanged ends (illustrated) is in nine standard pipe sizes from 2½ to 12 in., and standard lengths from 24 to 60 in. Type RHM (brass fitted couplings) is in eight pipe sizes from ¾ to 4 in., standard lengths from 12 up to 36 in.

Korfund Company, Inc., 48-53D 32nd Place, Long Island City 1, N. Y.

NOISE SURVEY METER

Palm-sized, to monitor and measure noise hazards, this noise survey instrument weighs 12 oz. It is designed for rapid, multi-location measurement of sound levels to establish noise-contour "maps" in large industrial plants.

Complete noise analysis may be achieved by using this meter in conjunction with the Soundscope, a precision instrument developed to measure the energy distribution of sound within a narrow frequency band.

The meter may be used either ahead of the Soundscope, as a means of locating areas requiring detailed study; or behind it, as a monitor of noise levels in areas where frequency characteristics have already been determined.

In making sound measurements, the survey meter is pointed toward the noise source, energized by battery

power, and adjusted to an "on-scale" reading by means of a single thumbwheel attenuator. Sound pressure levels may be read quickly in the range of 75 to 140 db.

Two models, the Industrial Hygiene model, designed with frequency response characteristics based on hearing damage criteria, and a second intended for use in noise reduction studies and acoustical engineering surveys, with a flat response, are offered.

Mine Safety Appliance Company, Pittsburgh, Pa.

ZONE VALVES

Thermostatically-controlled rotary type valves which may be easily installed in supply lines, in any position, are suitable for either new or existing hot water systems. Economically useful in multiple-zone hydronic heating systems, they eliminate the need for expensive flow valves, additional circulators or burner relays, of involved piping.

They are in three sizes, ¾, 1 and 1¼ in.

General Fittings Company, East Greenwich, R. I.

PIPE JOINT SEALER

In tape form, this joint sealer is made of Teflon and may be used with most industrial acids, corrosives, caustics, hydraulic fluids and aromatic fuels. Temperature range is from −250 to 500 F, pressures to thousands of pounds.

Known as Thred-Tape, it may be used on plastics, aluminum, stainless steel, ceramic, synthetic rubber and carbon pipe.

Self-lubricating, the sealer is in ½ x 288 in. rolls and is applied by wrapping tightly around the male threads. When the connection is made, the tape fills in the voids to provide a leak-proof seal.

Crane Packing Company, 6400 Oakton St., Morton Grove, Ill.

FILTER-DRIER BYPASS

As a compact assembly, this factory unit includes a replaceable cartridge type filter-drier, piping and valves which permit replacement of the molecular sieve cartridge without shutting down the refrigerant system.

Factory assembly results in uniform quality. Each one has a bracket, so that it may be mounted flush on a panel. The top flange may be removed without demounting the drier, as the bracket provides adequate clearance.

Line sizes of ⅜, ½ and ⅝-in. are offered at this time.

Remco, Inc., Zelienople, Pa.

(*Continued from page* 49)

E_s = shade transmission factor for solar calorimeter tests as calculated by individual coefficients, dimensionless

E'_s = experimentally determined shade transmission factor for solar calorimeter tests, dimensionless

ϵ_s = shade transmission factor for shade applied to a window, dimensionless

F_s = ratio of area of sunlit shade to total shade area, dimensionless

f_i = coefficient of heat transfer from inside surface of shade to room, Btu per (hr) (sq ft) (F)

f_o = coefficient of heat transfer from outside surface of cover glass or window to outside environment, Btu per (hr) (sq ft) (F)

I = rate of incidence of direct and diffuse solar radiation upon outside surface of cover glass or window, Btu per (hr) (sq ft)

k_G = thermal conductivity of cover glass or window, Btu per (hr) (sq ft) (F/in.)

Q_A = total energy transferred through shade absorbed by collector plate or room, Btu per hr

Q_1 = rate of incidence of solar radiation upon outside surface of shade, Btu per hr

Q_2 = solar radiation directly transmitted through shade, Btu per hr

Q_3 = solar radiation absorbed by outside surface of shade, Btu per hr

Q_4 = energy transferred by normal transmission from inside surface of shade to collector plate or room, Btu per hr

Q_5 = energy transferred by normal transmission from outside surface of shade through cover glass or window to outside, Btu per hr

Q_6 = directly transmitted solar radiation absorbed by collector plate, Btu per hr

t_i = interior temperature of room, F

t_o = outside temperature, F

t_p = temperature of the collector plate, F

t_s = temperature of shade, F

U = overall coefficient of heat transmission from outside to interior of room, Btu per (hr) (sq ft) (F)

U_{so} = coefficient of heat transmission from outside surface of shade through cover glass or window to outside, Btu per (hr) (sq ft) (F)

X_G = thickness of cover glass or window, in.

α_p = absorptivity of collector plate for solar radiation, dimensionless

α_s = absorptivity of outside surface of shade for solar radiation, dimensionless

ρ_G = reflectivity of glass surface for solar radiation, dimensionless

ρ_p = reflectivity of collector plate for solar radiation, dimensionless

ρ_s = reflectivity of outside surface of shade for solar radiation, dimensionless

$\rho_{s,B}$ = reflectivity of inside surface of shade for solar radiation, dimensionless

τ_G = transmissivity of cover glass or window glass for solar radiation, dimensionless.

τ_s = transmissivity of shade for solar radiation, dimensionless

ASHRAE OFFICERS, COMMITTEES
See page 76, this issue

STANDARDS COMMITTEE
See page 75, this issue

TECHNICAL COMMITTEES
See page 96, May JOURNAL

RESEARCH AND TECHNICAL COMMITTEES
See page 97, May JOURNAL

EMIG-SPAWN

(*Continued from page* 43)

Thermal stability has been studied by following weight loss at 160 C and 180 C. Fig. 5 shows the weight loss at 160 C compared with both yellow and black varnished glass fabric. Fig. 6 shows weight loss at 180 C as compared with a commercial polyester and two urethane coated glass fabric products.

Another property of the acrylic resin insulation, of likely interest, is its fast release of moisture which permits faster dehydration of the all complete systems in small or large hermetic equipment.

CONCLUSIONS

Acrylic resin coated magnet wire has excellent physical, chemical, and mechanical properties. Production tests have confirmed these results. Thermal stability data indicate such insulations possess adequate stability for operating at Class B temperature. Experimental and practical data on resistance to Refrigerant-22 indicate that acrylic resin insulation may be used in hermetic motors under more stringent conditions than present insulations and is being used with increasing acceptance for production in both hermetic and Class B applications.

Many of the products described herein and their uses are covered by U. S. Patents 2,787,561 and 2,787,603 and pending applications. The disclosures given in this discussion are not to be construed as conveying any patent rights.

The authors acknowledge the help given by P. F. Sanders and J. R. Huntsberger of the duPont Marshall Laboratory in the preparation of this paper.

For Performance — Capacity — Low Maintenance

Select a

LARKIN ZEPHYRCON

Air-Cooled Condenser

9 Belt-Driven Models—5 to 40 Tons 5 Direct-Drive Models—2 to 10 Tons

ENGINEERED FOR PARALLEL USE FOR VIRTUALLY ANY CAPACITY

Larkin Zephyrcon Air-Cooled Condensers are available in two series: direct-drive FC models—for economy and minimum maintenance; and belt-driven BFC models—for larger installations and the lowest possible noise level.

Multiple circuits—for operating more than one compressor from a single condenser—are available for all models at no extra cost.

PEAK PERFORMANCE YEAR 'ROUND

When used with the Larkin Posi-Trol control system, which maintains head pressures in low ambient temperatures, the Larkin Zephyrcon functions at peak performance the year 'round in any climate.

Compare the features point for point; check the performance data, and make the wise selection—Larkin Zephyrcon Air Cooled Condenser.

BASIC RATINGS (30°+D)

FC SERIES — DIRECT DRIVE

Model No.	*Capacity at Evaporator BTU/HR*	*Tons*	*Cfm*
FC-2	27,700	2.30	2500
FC-3	37,350	3.21	3500
FC-5	62,500	5.21	5500
FC-8	95,600	7.96	7500
FC-10	124,000	10.3	9100

BFC SERIES — BELT DRIVEN

Model No.	*Capacity at Evaporator BTU/HR*	*Tons*	*Cfm*
BFC-5	62,500	5.21	5500
BFC-8	95,600	7.96	7500
BFC-10	124,000	10.3	9100
BFC-13	149,200	12.4	12000
BFC-16	191,200	15.92	15000
BFC-20	248,000	20.6	18200
BFC-26	298,400	24.8	24000
BFC-32	382,400	31.8	30000
BFC-40	496,000	41.2	36400

OUTSTANDING FEATURES

- Patented Larkin cross-fin coil — aluminum fins and staggered copper tubes.
- For use indoors or outdoors.
- A single large fan on BFC model assures low operating noise level.
- Motor on BFC model is standard NEMA design mounted on adjustable base.
- Slow speed motor on FC model is permanently lubricated, has overload protection, and is mounted on resilient base.
- Casing is of heavy-gauge steel.
- Unit finished with baked-on open-base primer and melamine top coat for maximum protection.
- Motor is enclosed in casing, protected from the weather.
- Discharge shield available.
- Fan Guard is standard equipment.
- Return bends protected by heavy-gauge shield.

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With ASHRAE Chapters

(*Continued from page* 81)

selected a typical job as a basis for selecting condensing equipment. He used preset figures for water temperature, flow and gpm, choosing various types of condensing equipment to fit the conditions mentioned.

WEST TEXAS (H) . . . On the May program, Jack Roberts, Zumwalt and Vinther, consulting engineers, discussed some typical and special installations of air conditioning equipment used at the University of Texas, Southern Methodist University, and for the Computer Building of the Shell Oil Company, in Midland, Texas.

SOUTH FLORIDA (R) . . . Presenting his paper on the Application of Centrifugal Compressors to Water Cooling Duty, W. W. Salmond addressed this group meeting in April. He is the Southern District Manager for York Corporation.

In May, this group planned to hear David Rickelton, engineering consultant, Buensod-Stacey, Inc., on Design and Application of Dual Duct Systems.

LONG ISLAND (H) . . . John Engalitcheff, Jr., President, Baltimore Aircoil Company, planned to be with this group in May to cover Water Conservation Equipment – Theory and Application.

DAYTON (H) . . . Covering the uses and economics of electronic air filters and charcoal filters, Richard J. Looney, production Sales Manager for Air Cleaning Sales, Minneapolis-Honeywell Regulator Company, spoke here in mid-May. His topic was Air Purification.

ONTARIO (R) . . . Treating the basic theory behind the thermoelectric principle of refrigeration and the Peltier Effect with a humourous note, James K. Nelson, Westinghouse Electric Company, met with these members in April.

The next month's meeting was preceded by a plant tour of the Hussmann Refrigeration Company, Ltd. The speaker for the evening was Parker V. Phillips, Service Manager, Hussmann, who chose as his topic, the Relationship between the Field Service Engineer and the Design Engineer.

LOUISVILLE and BLUEGRASS . . . Industry's demands for smaller size, lighter weight, lower cost, higher speeds and capacity modulation in compressors present a real challenge in maintaining compressor life, Clayton B. Cramer said, at the April joint meeting. Mr. Cramer, who is Assistant to Manager, Compressor Dept, Carrier Corporation, addressed the members on Why Compressors Grow Old.

He mentioned some of the common misapplications of the units. Overloading by changing re-

frigerants (as much as 57% when changing from Refrigerant 12 to 22) was one, in proper suction pipe size for proper oil return was another. At least 1400 fpm velocity is required in this case, he reminded his listeners. Conditions causing dilution of refrigerant in crankcase oil was a third case in point. Some conditions, he noted, yield 1.5 lb of refrigerant for each 1 lb of oil.

Effects of moisture and other system contaminations of compressor life also came under scrutiny.

Later in the evening's program, Louis H. Fitzmayer, Radiation Engineer, General Electric Company's Range Dept, presented a brief summation of the electronic oven. High initial cost is the present retarding factor in the electronic oven market, he said.

Also meeting jointly in May, these members planned to hear D. A. Boehmer, Sales Dept, Republic Steel Corporation, document his films of Russia, taken while on a recent trip.

CENTRAL MICHIGAN (R) . . . Chief Engineer, Automotive Air Conditioning Dept, Chrysler Corporation, Joseph Loveley planned to address this chapter in May on design and application of automotive equipment.

MEMPHIS (H) . . . In a study made by Minneapolis-Honeywell Regulator Company, percentage cost of complete air conditioning was related to the increased efficiency needed to justify it. William J. Ortman, Commercial Div, Market Manager for Industrial Buildings, Minneapolis-Honeywell, explained this study to chapter members meeting in April. His talk was entitled The Economics of Year-Round Air Conditioning.

SAN JOAQUIN (R) . . . Branch Manager of the Fresno office of Minneapolis-Honeywell Regulator Company, Floyd Stroup, spoke on Basic Controls as Applied to Air Conditioning, at the April meeting of this chapter.

NATIONAL CAPITAL and WASHINGTON, D. C. . . . Application and cost data on both single state and compound system heat pumps were the topics of R. J. Augusterfer, York Corporation, at the April meeting, a joint one.

JOHNSTOWN (H) . . . Relative merits of ground source, water source and air source heat pumps were the emphasis at the April meeting when R. D. McNatt addressed the group. The air source heat pump received his particular attention as it is the most popular. The speaker, Pittsburgh Branch Manager, York Corporation, gave reasons for using different types of compressors and refrigerants, and reviewed the design of air to refrigerant heat exchangers. He also covered the economic advantages and limitations of commercial heat pumps, with respect to several recent applications. The effect of the two-stage heat pump and packaged systems on the economic picture was noted.

NEW COMPLETE LINE OF HONEYWELL ACTUATORS AND DAMPERS FOR YOUR EVERY NEED!

Honeywell's new M630 Actuator motor.

Honeywell's D42 Damper

Honeywell Actuators—to position dampers, valves, pre-rotation vanes, skylights, defrosting doors—any application up to 750 pounds of torque

Honeywell can help you satisfy an almost unlimited number of power positioning requirements. You can choose from 11 models which offer: torque ratings from 25 to 750 inch pounds; timing ranges of 7.5 seconds to 4 minutes; 3 control methods—two position, proportioning and floating; 2 drive types—reversible with or without spring return and/or unidirectional.

Other outstanding features of Honeywell's Actuator line include: choice of tandem or sequence switching; 360° adjustable crank arm; ease of mounting and adjustment; quiet, oil-geared operation; sturdy, durable Honeywell components for years of trouble-free operation.

Honeywell, the biggest name in dampers for heating, ventilating, air conditioning and specialized applications

The custom-built features of Honeywell motorized dampers give them an almost limitless versatility. In addition, they give instant response to controllers along with an assured tight seal. They have extra heavy bracing to assure true alignment and consistent accuracy of positioning. Operation is noiseless—and trouble-free.

The custom-built features of the Honeywell Damper line also mean easier ordering, simplified installation. Get complete and detailed information on Honeywell's complete line of dampers and actuator motors by calling your local Honeywell office. Or write Honeywell, Dept. RE-6-20, Minneapolis 8, Minnesota.

Honeywell

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Applications

CHILLED WATER SYSTEM FOR METALLURGICAL LAB

Nearing completion, a new metallurgical laboratory at Allegheny Ludlum Steel Corporation's Brakenridge, Pa., research center will use a hermetic centrifugal refrigeration machine as the cooling source for air conditioning. The two-story structure, built of stainless steel, glass and masonry, will provide 62,000 sq ft of additional research facilities when completed later this year.

The compact, completely integrated Carrier Corporation centrifugal, with a cooling capacity of 338 ton, will be located in a separate powerhouse. The electrically-driven unit will supply chilled water at 42 F through underground pipes to air handling equipment on the roof of the laboratory. A dual duct system will distribute air to the conditioned area.

MOTEL-ICE RINK COMBINATION

Air conditioning a 250-room motel in the summer, and supplying brine for a skating rink in the winter, two Acme Industries combination packaged liquid chillers are inter-connected with a special cooling tower. The system uses dual refrigerant feed control, electric control panels and an automatic change-over switch for temperature-pressure controls.

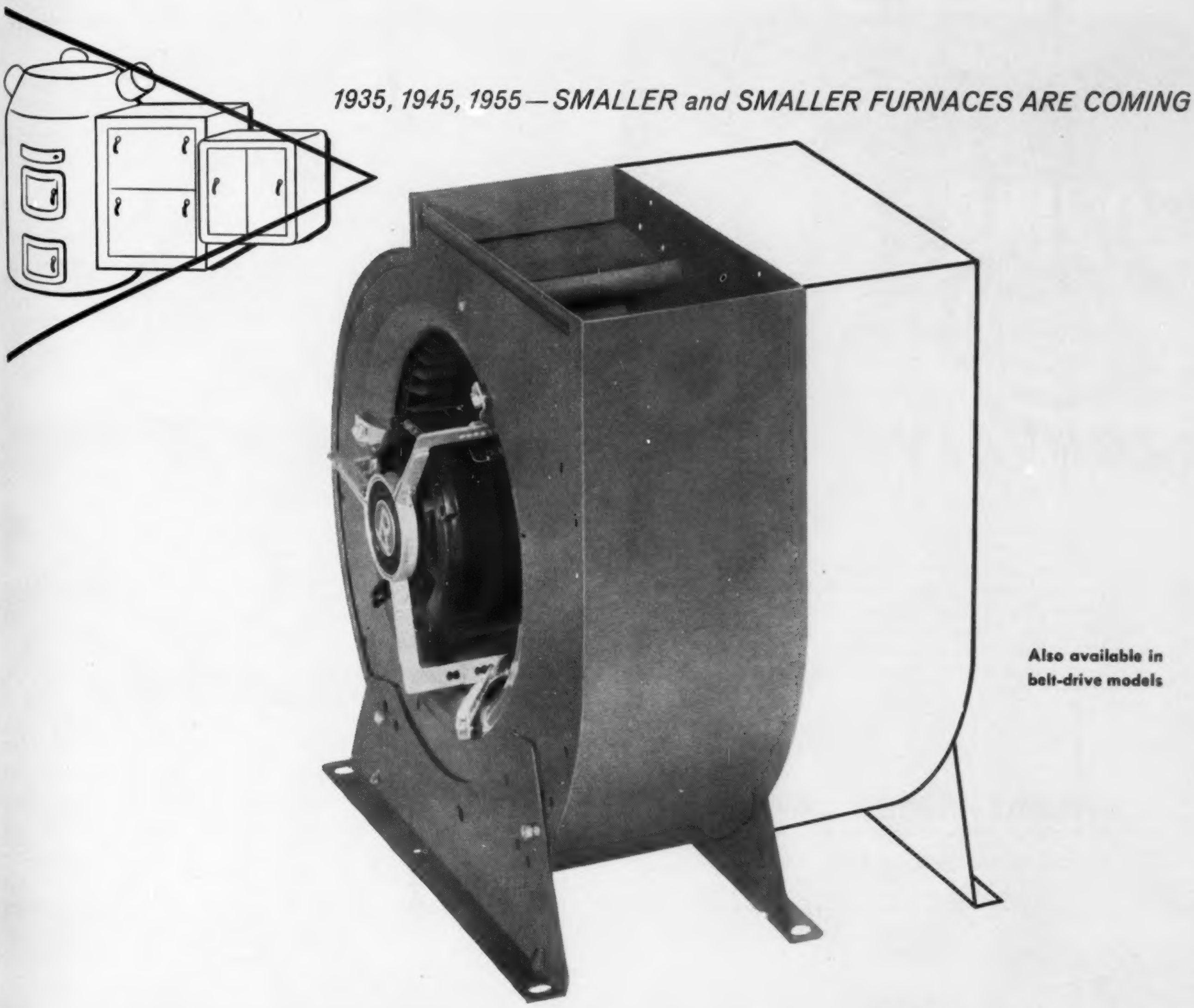
COMPLEX AIR DISTRIBUTION SYSTEM FOR 22,500-SEAT SPORTS ARENA

Air conditioning in a structure as large as the new Los Angeles Memorial Sports Arena is necessarily complex, as it must provide for a wide range of seating conditions. The ceiling of the elliptical structure is supported by a span of steel, eliminating inside supporting columns.

The ice rink is serviced by four packaged water-chilling plants, two of which may also chill brine.

The air distribution system for the 22,500-seat arena was designed to serve the arena floor, concourse and seating area independent of one another. For the permanent seating area there are eight central fan units located in an overhead furred soffit. Each unit is designed to deliver 50,000 cfm of air. Individual units are equipped with outdoor and recirculation air dampers, combinations return fans, exhaust fans and dual temperature heating and cooling coils. The piping system supplying these units is arranged as a two pipe reverse return water system providing chilled or hot water as required.

The arena floor area is served by six Recold Corporation air handling units also located in the furred soffit. These supply either heating or cooling. Each unit is thermostatically controlled to the temperature of the area it serves. The concourse floor of the



THE NEW BRUNDAGE *SLIM* 'B'
SHAPED FOR TOMORROW'S SALES

Within 5 years, it has been predicted, 89% of the furnaces sold will be 75,000 BTU or less. Your own experience with smaller and smaller sizes makes the trend clear.

The Brundage Slim 'B' fits this pattern as no other blower can. Using any of the Slim 'B' sizes, you cut blower space requirements by ⅓ to ½ . . . and achieve designs that fit your future sales.

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DOING JUST 1 THING BETTER SINCE 1919

Memorial Sports Arena is considered as four separate zones with each of these zones having its own air handling unit capable of supplying either cooled or heated air. Ventilation of the individual systems is normally controlled thermostatically, but a central station control allows prompt correction if any area needs more or less air tempering.

TWO WATER TEMPERATURES OFFERED IN HOTEL INDUCTION SYSTEM

Complete control of the temperature in his hotel room is afforded the Statler-Hilton guest in Boston in a

recent York Corporation installation. The Three-Pipe Hi-I air conditioning and heating system derives its name from the hot water, cold water and return pipes that connect to the high pressure induction unit in each room.

Both cold and warm water are always available to each induction unit control valve to provide either heating or cooling. A conventional chilled water supply and return system is supplemented by a small warm water supply line.

The automatic control valve does not mix warm and cold water, but selects varying quantities of either, not both, and modulates the flow to the induction unit coils to maintain the desired room temperature. Seasonal changeover is eliminated.

Primary air is delivered from a central system to each induction unit. Duplication of secondary water pumps is not required; there is one secondary pump for warm water and one for chilled.

This system is a recent development. It was first introduced to the industry at the International Heating and Air Conditioning Exposition in Philadelphia last January.

GAS-FIRED RADIANT HEATERS ELIMINATE CONVEYOR TIE-UP

Sheet aluminum with integral tube circuits, "Tubed Sheet" of Reynolds Metals Company manufacture, is made by printing a reduced-size circuit pattern on one sheet and bonding another to it in a rolling mill which also expands the size of the sheet and the pattern four times. Metal ink is used to prevent bonding of the printed surfaces which are expanded into tube circuits by fluid under pressure after rolling is completed.

Actual operation of this new system at mass pro-

duction rates revealed that more radiant heat was required than was provided by the ten-bank, 200 kw system originally installed. The printed surfaces were smearing when the second sheet was placed on the first.

By substituting a single bank of ten Perfection-Schwank gas-fired infra-red heaters, it was possible to provide a bank of 480,000 Btu/hr capacity over 22 ft of conveyor. Each gas unit has a capacity of 48,000 Btu/hr.

The units convert 60% of fuel input into infra-red rays of lengths (1.5 to 6 micron) readily absorbed by the metal ink. This range of infra-red wave length is generated by an element surface temperature of 1600 F. The ceramic is perforated with 200 small holes per sq in. Air drawn from the atmosphere and gas are mixed in a chamber behind the ceramic and passed through the small holes. A separate flame burns at each hole and the entire combustion takes place within ⅛ in. of the ceramic face. The rear surface of the ceramic does not exceed 400 F.

Heaters are ignited by a pilot light and protected against escaping gas by a pilot-heated thermocouple which closes an electrical solenoid valve in case the pilot fails.

SPACE SAVING SYSTEMS HEAT, COOL U.S. SHIPS

High velocity air conditioning equipment has already largely replaced conventional wide duct systems on ships constructed in European yards in recent years. This advanced type system is now in operation on the American President Line's **S. S. President Garfield.**

Small diameter pipework of the HiPress Air Conditioning of America, Inc. delivers either heated or cooled air upon demand to individually operated room units. The twin-pipe systems with air velocities of 5000 fpm eliminate the fire hazard of large-duct work.

The **President Garfield's** air conditioning plant serves a total of 62 individual spaces throughout the ship, including public rooms and crew members.

COUNTERFLOW CONTINUOUS POULTRY CHILLER

Fifty birds per min are processed in a system using refrigerated water and a stainless-steel flume equipped with a conveyor.

The poultry, at the Holly Farms Poultry Processing Plant, enters one end of a 54 ft tank at a temperature between 80 and 95 F. Cold water, at 32 or 33 F enters at the opposite end and is not recirculated. The poultry is conveyed in counter-current fashion against the stream of refrigerated water by fingers or rakes. After moving through the cold water for about 1½ hr, the birds are lifted out of the flume by the fingers, at a temperature of 36 F.

The fresh water is chilled in stages by means of high-efficiency coolers combined with an ice reserve unit, which uses off-peak power. The ice builder, by storing cold, takes care of surges in the load, and

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provides extremely cold water. Continuous operation of the chiller with the aid of conveyors replaces the batch system.

The water in the Frick Company chiller is kept fresh and clean by the one-way counter-flow movement. Debris is removed immediately by the washing action when the poultry first enters the water.

PARTITIONING FREEDOM GAINED WITH LOW SPEED AIR FLOW

Uncluttered ceilings and freedom of partition movement have been achieved through the use of low velocity air diffusion in a new modular-designed industrial project. The low velocity diffusion system was specified for the air conditioning of the new Sikorsky Engineering Building to gain space flexibility.

For the new quarters, 5100 Multi-Vent Air Diffusers of the Pyle-National Company have been installed in the acoustical metal pan ceilings of office, general and engineering space. The fixtures consist of flush-fitting modular diffusion panels connected by flexible tubing to the duct system.

By means of a valve arrangement within the panel, the air is slowed to a 30-40 fpm flow and dropped into the room vertically from a 2 x 4 ft perforated plate.

HOSPITAL AIR CONDITIONING EFFECTED IN 24 HOURS

One-day installation of a complete air conditioning system at New York's Mother Cabrini Hospital proved that interruption is not always necessary. The rooftop type installation was possible through use of Dunham-Bush, Inc. package chillers with pump, air handling unit and compressor.

METAL-PLASTICS BARRIERS

Aluminum and plastics film laminates form moisture vapor barriers for a Grand Union Perishable Foods Warehouse joint project of Acme Backing Corporation and Johns-Manville Corporation.

Wall sections of the over 4 million cu ft of refrigerated storage space were constructed by the horizontal method. The insulating materials were adhered to these rock cork sections before they were hoisted into position in the wall.

Where moisture vapor barrier materials were used under floors, they were laid in the same manner, prior to the pouring of the concrete. To permit adjustability for thermal expansion and contraction, accordion-pleated strips of Acmeflex material were cemented to the wall sections.

Moisture barrier material is especially useful in ripening rooms where humidities of 95% and temperatures approaching 100 F are used. Selected concentrations of ethylene oxide gas, injected to help ripening, are maintained by the barrier as the permeability of the gas is close to zero.